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THE  
**STEAM-ENGINE**  
AND  
**OTHER STEAM-MOTORS**

*A TEXT-BOOK FOR ENGINEERING COLLEGES  
AND A TREATISE FOR ENGINEERS*

IN TWO VOLUMES

VOLUME TWO  
FORM, CONSTRUCTION, AND WORKING OF  
THE ENGINE; THE STEAM TURBINE

BY  
ROBERT C. H. HECK, M.E.

*Assistant Professor of Mechanical Engineering  
Lehigh University*

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**SECOND ISSUE**

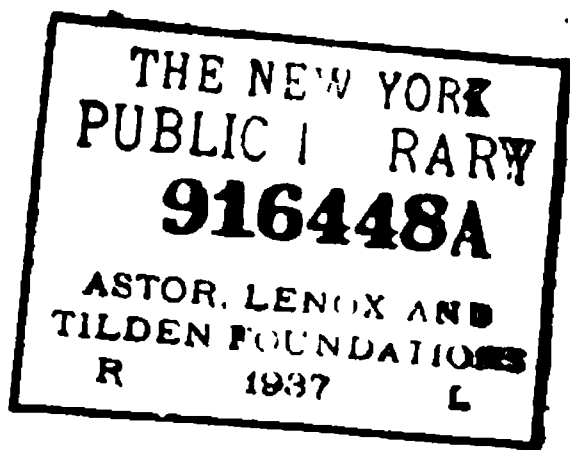
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## PREFACE.

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IN the effort to attain the objects set forth in the preface to Vol. I., the share of the present volume is chiefly "to give a broad description of constructive practice and to explain fully the working of the machine in its several departments." The steam-turbine is included, but with a much less extensive treatment than is given to the pressure-engine. A proposed chapter on accessories has been replaced by one on steam-engine performance, in which the results from a large number of the best and most reliable tests of engines and turbines are tabulated and discussed. The aim is to cover broadly and thoroughly the engine as a machine and as a member of the power-plant, but not to take up other phases of the subject of power-plant engineering.

The collection of photographic illustrations in the introductory section, largely from builders' catalogues, is intended to give a comprehensive knowledge of the general form of the engine, in a shape always convenient for reference. But in the description of constructive detail line-drawings are used almost exclusively: nearly all of them have been made expressly for this book, thus affording great freedom in the selection of examples, while facilitating adaptation to the space available, and permitting especial emphasis to be laid upon the essential point in the particular connection.

In the illustration of each department—of main engine and of subsidiary mechanism—the idea has been to choose typical examples, which will show various ways of meeting the problem presented. Where there is much complexity, the first case is very fully described; thereafter only the salient features are touched upon. Most of the detail sectional views are cross-hatched for material, according to the conventions in Fig. 199.

The chapters on *valve-gears* and *governors* treat their subjects from the two *view-points* of *description* and of *mechanical analysis*, combining into one discussion the two parts which for the main engine mechanism are separately set forth in Chapters VII. and VIII. The wide differentiation among designers causes a fairly complete presentation of *valve-gears* to cover a great deal of space. In the *governors*, analysis of several actual examples is used to establish the mechanical principles, but no attempt is made to reduce the design problems of design to mathematical expression.

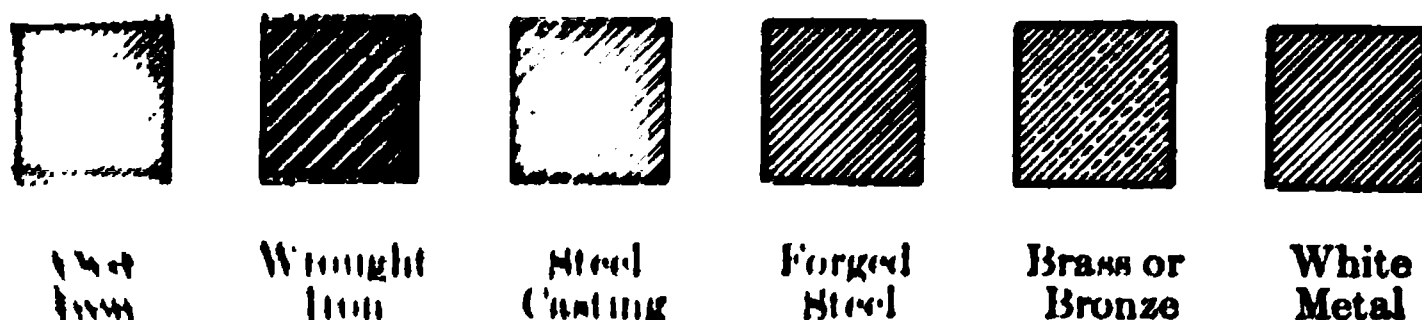


FIG. 100. Conventional Cross-hatchings.

The discussion of steam-action in the compound engine at first carries the simple pressure-volume analysis to the limit of utility; then a quantity of practical information is given, in the form of *area* steam diagrams and of realized cylinder-proportions. These diagrams help to illustrate, and have an important bearing upon, the question of economical performance, as considered in the last chapter.

Under the steam-turbine are presented, first, the general form and action of the important types; next, the simplified mechanics of the jet action; third, a discussion of the secondary influences which modify the simple, ideal case; last, a number of examples of constructive detail. As regards the difference in amount of space occupied, the idea has been to give in regard to the engine all the information that the student need seek from a formal treatise, thus far exceeding the bounds of a text intended wholly for class-room use, but to the turbine is given merely a text-book treatment of the fundamentals. This is a new and rapidly developing subject, and the object sought here is to lay a foundation for the wider knowledge to be got from special works.

The last chapter, on thermodynamic performance, is the fruit

of a careful selection from the great mass of information along this line contained in the technical literature of the last thirty years. More than two hundred tests are gathered into the principal tables, representing over half as many individual engines and turbines. Comparisons showing the effect of changes in the governing conditions have been warmly welcomed; but series runs, with the load as chief variable, are left for graphical representation, by a number of sets of performance-curves. Besides setting forth the essential observations and results, the tables give the thermal quantities involved, so that all the machines are first brought to a common basis of actual or absolute thermodynamic efficiency, and are then compared with the ideal case of the Rankine Cycle. On account of the great variety in character and fullness of the data, engine-efficiency rather than plant-efficiency has been given and emphasized. Lack of space has excluded many secondary data which would be of considerable interest, but are not of the first importance. The purpose has been to concentrate information which is otherwise widely scattered, presenting the more important facts and relations, and giving to the student and engineer a comprehensive knowledge of what has been done, upon which to base judgment of present performance and expectation for the future.

It has not been thought desirable to burden the text with references showing the source of every illustration. Many drawings have been made from makers' prints and advertising literature; the majority, perhaps, have been redrawn from engineering periodicals, usually with the selection of only the part desired; a few have been copied from other books, notably Leist's *Steuerungen für Dampfmaschinen* and Stodola's *Steam Turbines*, Third Edition (German); while half a dozen line-drawings from catalogues have been transferred directly. As to periodicals, *Engineering*, *Power*, *The American Engineer and Railroad Journal*, and *Marine Engineering* have been freely drawn upon for material.

R. C. H. H.

SOUTH BETHLEHEM, PA., March, 1907.

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## CHAPTER VIII.

### FORM AND CONSTRUCTION OF THE ENGINE.

#### § 41. Classification and General Form.

(a) PRIMARY DIVISIONS INTO CLASSES.—Elaborating somewhat the classification already set forth in § 4, and still looking at the subject from the side of adaptation to different conditions of service, the most obvious and useful division—remembering that no hard and fast lines can be drawn—seems to be as follows:

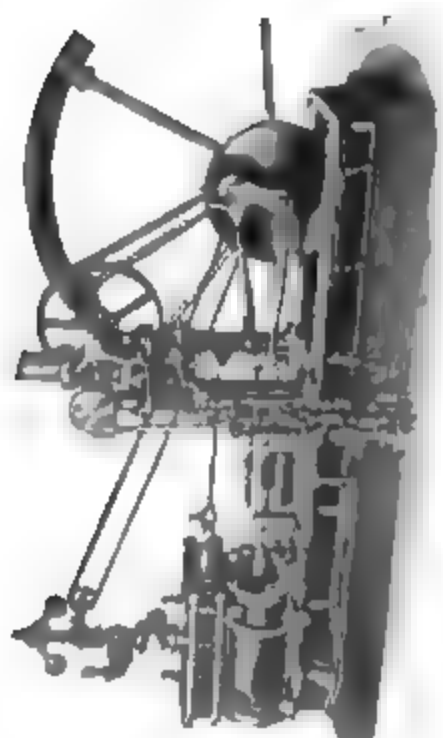
1. *Stationary Engines for the Generation of Power.* In every case, the engine has a rotary load, or the power is delivered through the shaft. This power may either be transmitted mechanically or be first changed into electric current.

2. *Directly Loaded Stationary Engines.* The working machine, other than an electric generator, is closely and intimately connected with the engine. Power may be delivered through the shaft, as in a mine-hoist or a rolling-mill; or through the piston-rod, as in steam-pumps and air-compressors. For direct connection to the electric generator, so little change from the simple belted arrangement is required that it seems better to include engines for that service under the first heading.

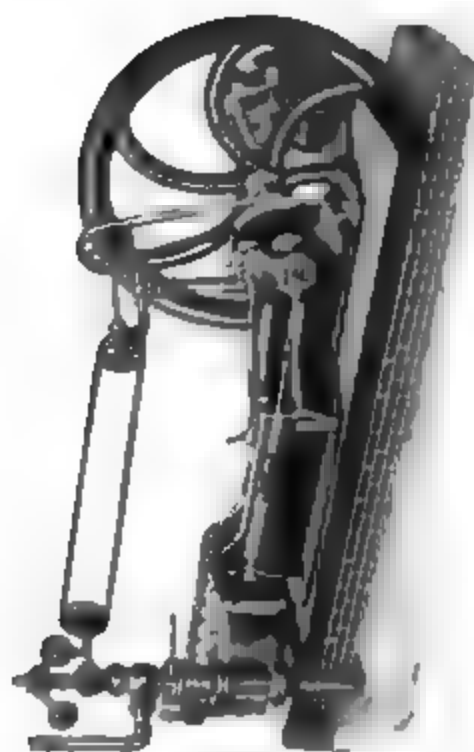
3. *The Locomotive.* With many variations in detail, this conforms closely to one prevailing general type.

4. *Marine Engines.* These, if limited to modern designs for driving screw-propellers, differ little in essential form. Going back to earlier practice, and including the less important class of paddle-wheel steamers, we encounter a wide variety in type, and some very peculiar forms.

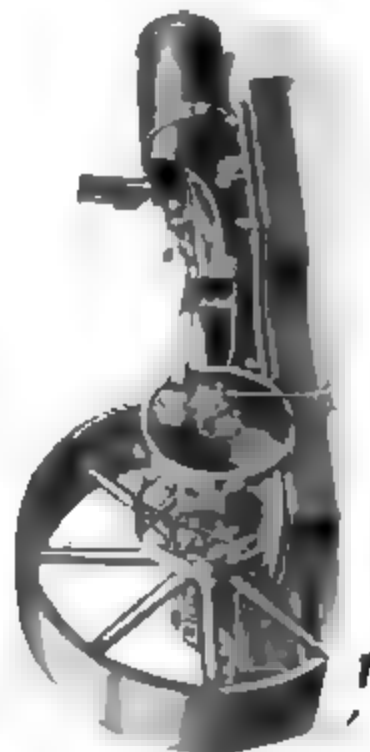
5. *Miscellaneous.* A class with this title is needed for the minor special types, for good designs that have been superseded in the



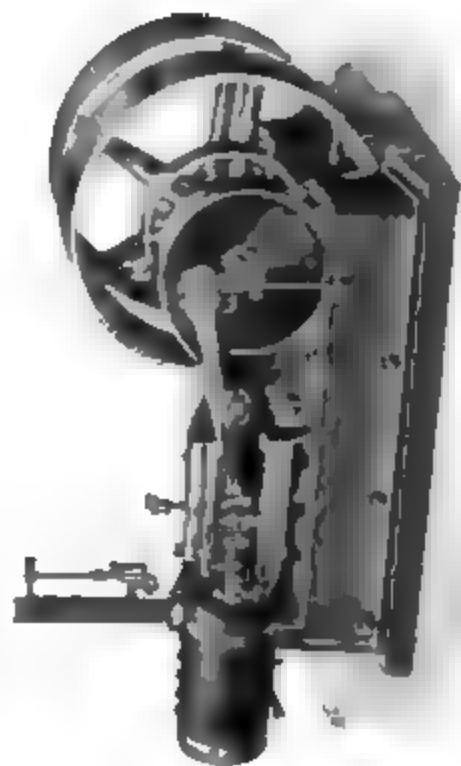
I. 1850-1860.



II. 1860-1870.



III. 1888-1900.



IV. 1884-1900.

FIG. 200.—Evolution of the High-speed Engine. Buckeye Engine Company.

course of evolution, and for some peculiar designs which just escape being freaks.

It will be noted that only the piston-engine is covered by this classification.

(b) STATIONARY ENGINES.—The greater number of the examples here illustrated by photographic views are taken from this widely extensive field. Subclasses which suggest themselves are:

1. *The High-speed Engine*, characterized by high rotative speed, not carried to very large sizes, and especially adapted to running electric generators.

2. *The Corliss Type*, developed as the standard factory engine, and in its smaller sizes used mostly for driving machinery. Many engines besides those following the Corliss design have the same fundamental characteristics.

3. *The Power-house Engine*, for direct connection to large generators. There has been, along parallel lines, an enlarging and speeding up of the Corliss engine, and a development of the high-speed type to large sizes, with the adoption of some ideas from marine practice.

(c) EVOLUTION OF THE HIGH-SPEED ENGINE.—This is very well shown by the group of engines in Fig. 200. In I., the form of the bed-plate is the first noticeable feature: it is a plain rectangular cast-iron frame, with the cylinder, the guide-bars, and the main bearing bolted to it. Evidently modelled on the wooden framework which preceded it in the crudest construction, it might, in many designs contemporary with this, have served almost equally well for the bed-plate of a lathe.

In this engine, the cylinder has brackets cast on the sides of it, which rest upon the bed-plate: and the steam-forces are resisted, if moderate by the friction due to the grip of the bolts, if severe by shearing stress in the bolts. Except in the locomotive, an end-connection between cylinder and frame is now used; it is more directly in the line of force and is easier to fit truly; while simple tension in the bolts is decidedly preferable to pin-action.

At II. we see a long step forward in the direction of adapting the engine-bed to the particular functions which it has to perform—namely, to carrying the cylinder, the cross-head guides, and the

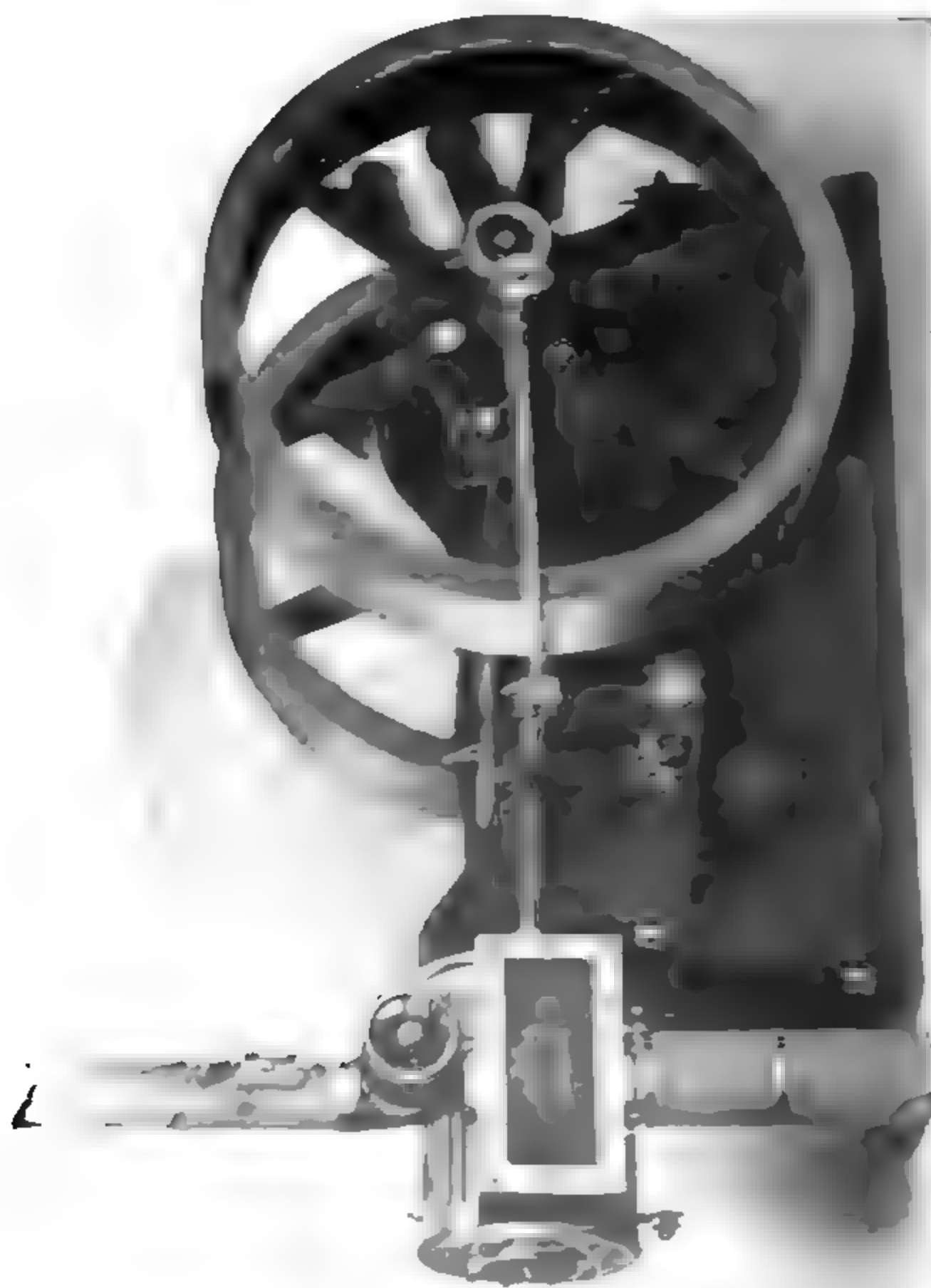


FIG. 201. American Ball High-speed Engine, American Pacific Company.

shaft-bearing, and to resisting the working-forces in the machine. But the engine still has the plain slide-valve, and is governed by throttling the steam. Note that in both II. and I. the valve is driven by a return-crank, from the main crank-pin, instead of by the more usual eccentric on the shaft, behind the main bearing.

The engines at III. and IV. show fully developed types: III. has what is known as the Tangye bed, from the builders, an English firm, who introduced it. Near to the cylinder, the bed is given a cylindrical form; then it is cut away by a curved intersecting surface, so as to sweep down in graceful lines on each side of the guide-bars. IV. differs from III. chiefly in the form of the bed at the guides; in effect this is a part of the Tangye bed turned on its side, and a different type of cross-head becomes necessary. In both cases, the main bearing is, of course, a part of the frame-casting.

(d) ENGINES WITH THE TANGYE BED.—One good example of this style of design has been fully illustrated in Figs. 2 to 7; another is given in Fig. 201, and is further shown in detail by Figs. 237 and 251. In both engines, the sides of the bed are kept well above the horizontal plane through the cylinder-axis, so as to have a good part of the metal in the line of the principal working-forces. The moving parts are covered in by suitable oil-guards, to prevent the throwing of oil. In the Ames engine, Figs. 2 and 5, the method of lubrication by separate oil-cups is used. On the engine in Fig. 201 there is an oil-pump, driven by the rocker-arm and partly appearing in the photograph; from this pump, pipes run to all the bearings, with a sight-feed drip, adjustable as to rate of flow, at each one. On Fig. 202 is seen the gravity system, consisting of a central reservoir and pipes to all the bearings, with the sight-drips right under the oil-tank where they can be most easily watched.

In viewing this vertical compound engine, Fig. 202, we are now concerned chiefly with the general, external form, leaving mechanical detail and steam-action to be taken up later. Beginning at the bottom of the framework, the first part is the sub-base, which carries both engine and dynamo: then, for the engine proper, we have the bed-plate with the bearings, the cast-iron upright or housing at the back, and the steel braces or stanchions in front. The

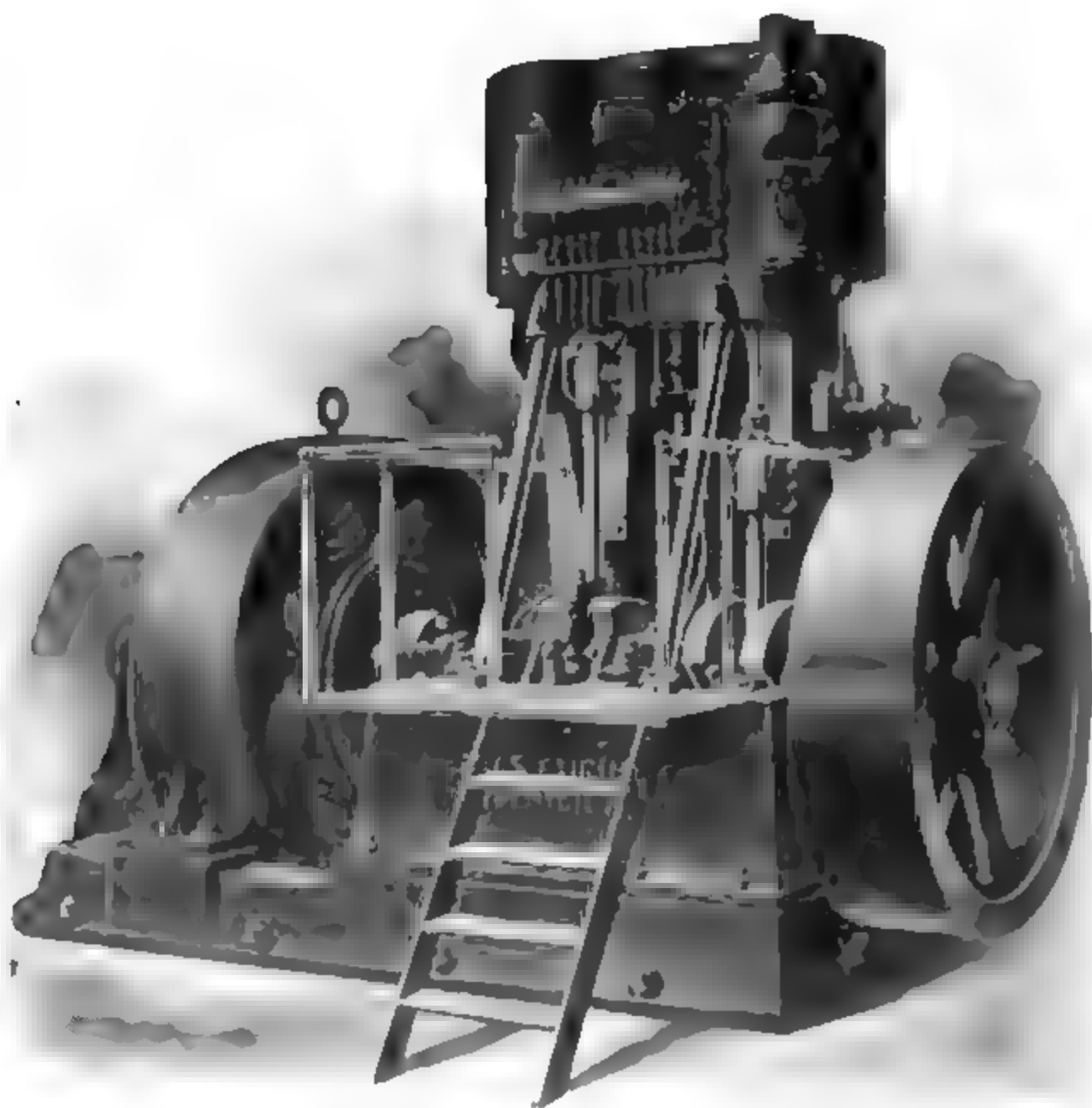


FIG. 202. - The Reeves Vertical Compound Engine.



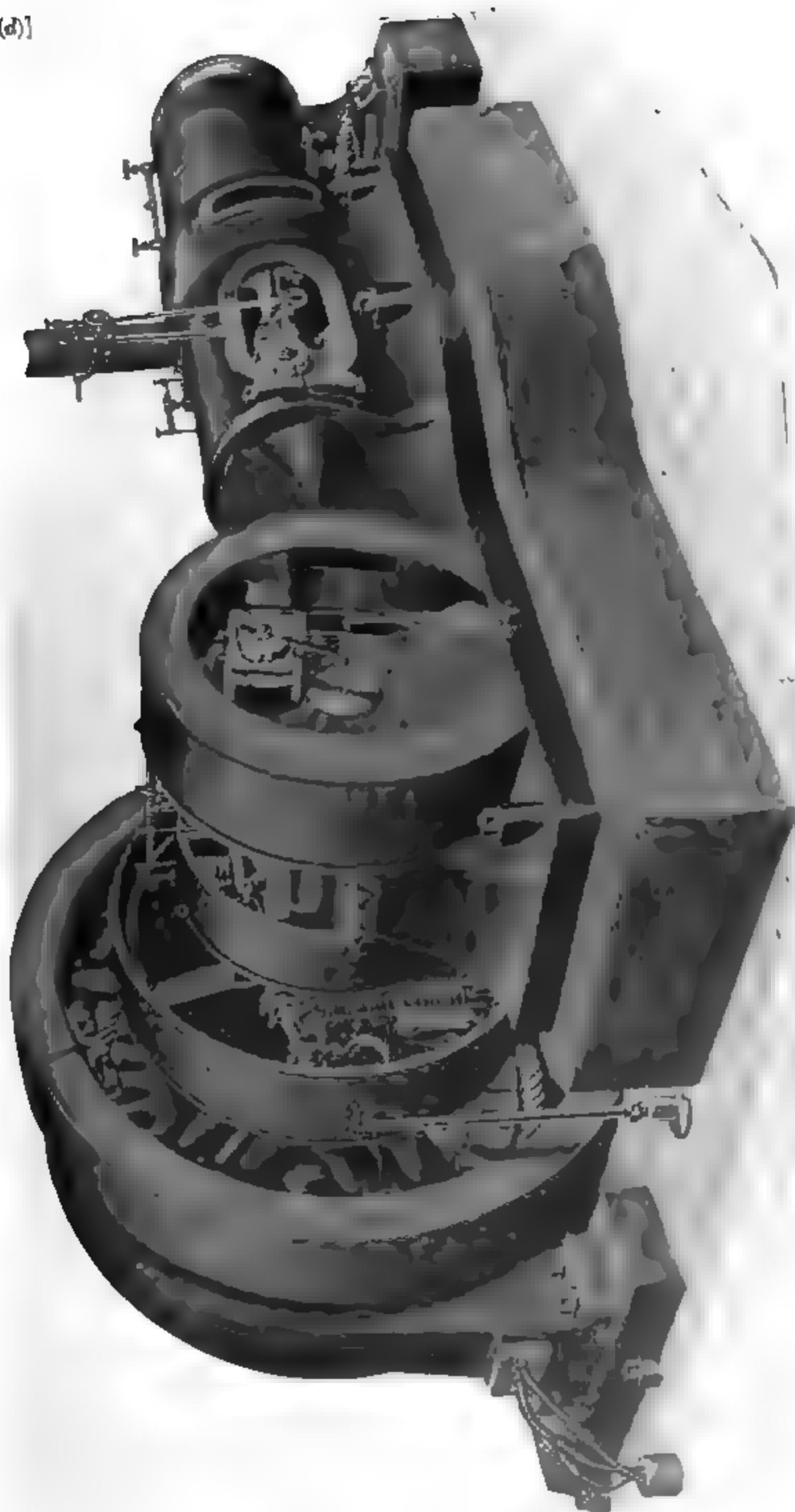


FIG. 203. Heavy-duty High-speed Buick Engine.

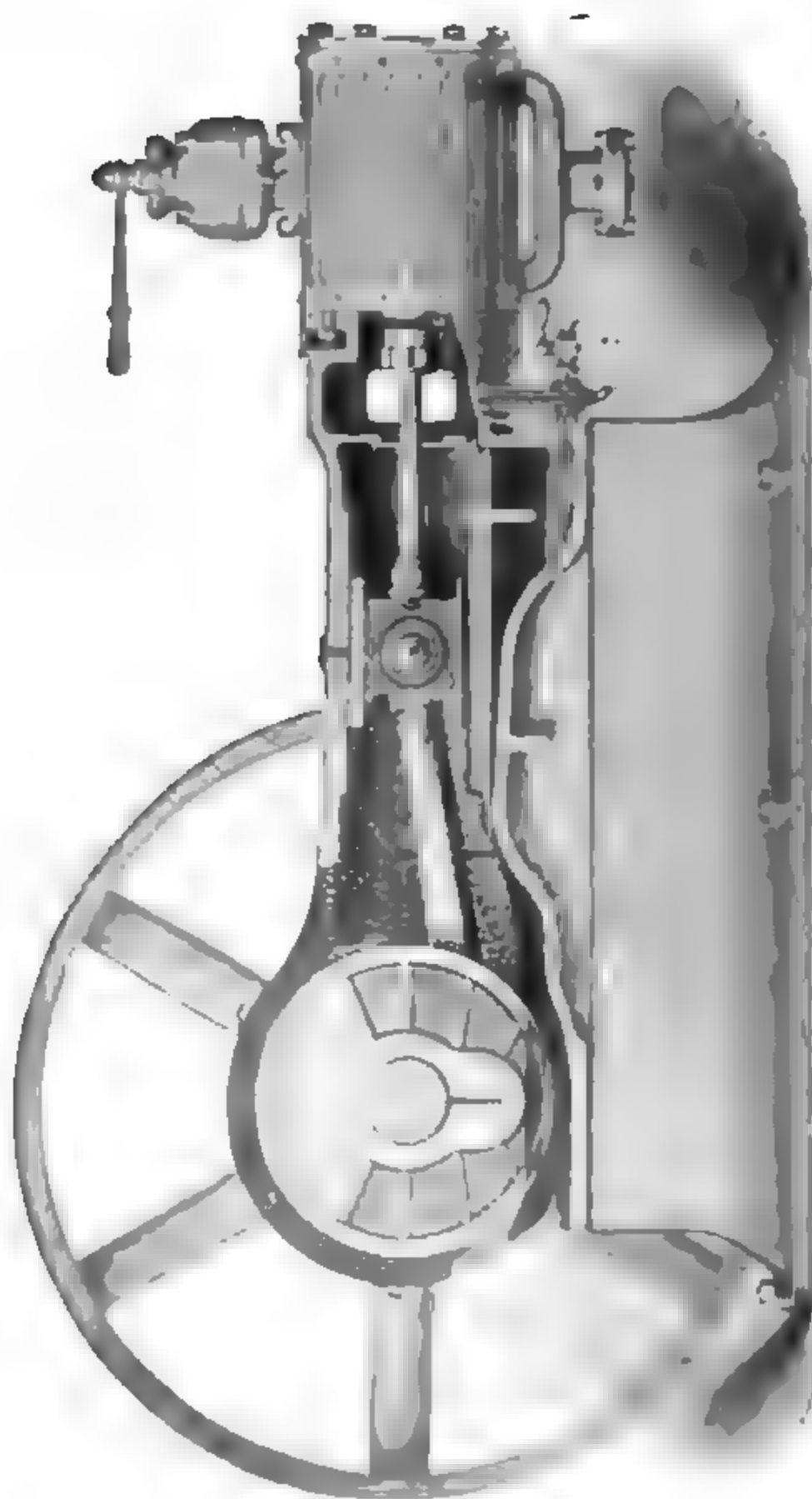


FIG. 204.—Section of Fig. 205, showing lubrication.

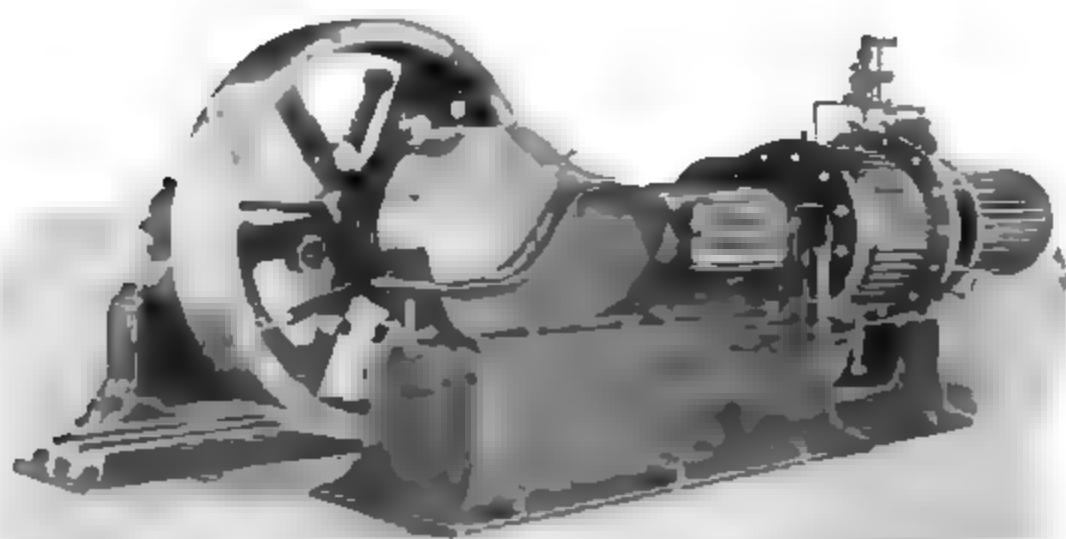
frame follows closely the Tangye outline near the cylinder; but at the shaft-bearings there must be a radical difference in form between vertical and horizontal engines. Like most stationary vertical engines, this differs from the marine type in that the frame is complete in itself, with a full seating upon which the cylinder rests; whereas in marine engines the cylinder is a part of the frame, the



FIG. 205 An Enclosed Center-crank Engine.

front and the back housings or columns being bolted to it, and not usually otherwise connected to each other at the top. This compound engine is of the direct-expansion type (see § 22 (c)), having its cranks opposite. When ready for running, the working-parts are fully enclosed by oil-guards of sheet-metal.

(c) ENGINES WITH ENCLOSED GUIDES. Another typical form of engine-frame is shown in Fig. 203. Here the guiding surfaces for the cross-head are parts of a cylindrical surface, so that they



I Front View



II. Rear View.

FIG. 296.—Tandem-compound Engine, Harrisburg Foundry and Machine Works.



can be bored out when the cylinder-seating is faced, thus insuring true alignment. This gives the frame a cylindrical form at the guides, and the latter are completely enclosed, except where openings are made at the sides for access to the moving parts. From one side of the guide-barrel, the frame is carried forward to the main bearing; and its outline, viewed from above, suggests the name "bayonet-frame" sometimes given to the type—more especially when it is used for long-stroke engines, not so compact as this one. In front of the guides and around the crank-disk is formed a casing of sheet-metal to catch oil, this engine having a gravity oil-system, but with the central oil-tank usually placed in some less conspicuous position than just above the engine. The sub-base is of concrete, rising from the main foundation: and the support under the cylinder is a rather unusual feature in a simple high-speed engine.

In Fig. 205 is shown an engine in which the idea of completely enclosing the moving parts is very fully carried out. The frame is box-shaped at the guides, then widens and forks to the bearings, with a separate crank-case tightly covering the crank-disks. The builders of this engine were the pioneers in the use of splash-lubrication, illustrated by Fig. 204, and now used in many other designs. Oil from the reservoir under the crank is continually thrown about the enclosed space: the guides are lubricated directly and very copiously; for the shaft-bearings, oil is caught in narrow troughs on the sides of the crank-case, from which it flows in steady streams; escaping from the inner ends of the bearings, the oil gets into recesses formed around the shaft on the outer faces of the disks, and is carried by centrifugal force to the surface of the crank-pin, through diagonal holes drilled as shown by the dotted lines on Fig. 5; oil for the wrist-pin is caught in grooves on the top of the cross-head—though in some recent designs a more positive feed is secured by putting the wrist-pin on the same system that supplies the main bearings. By this method a rapid circulation of oil is secured, with full supply to all the bearings, and with a minimum amount of attention.

Fig. 206 shows two views of another design where the same lines of construction and operation are followed, but in a side-crank

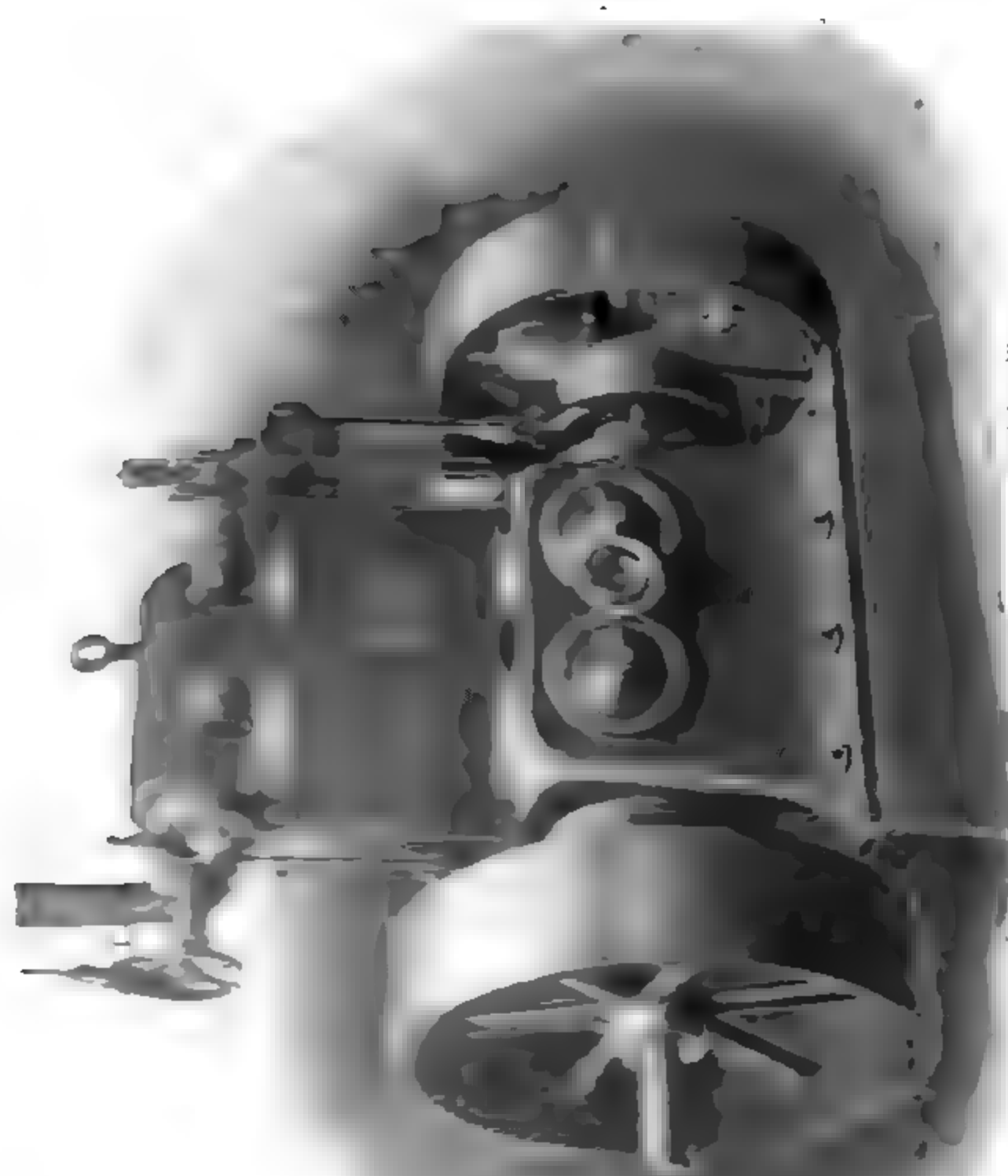


FIG. 207. Westinghouse Single Acting Compound Engine

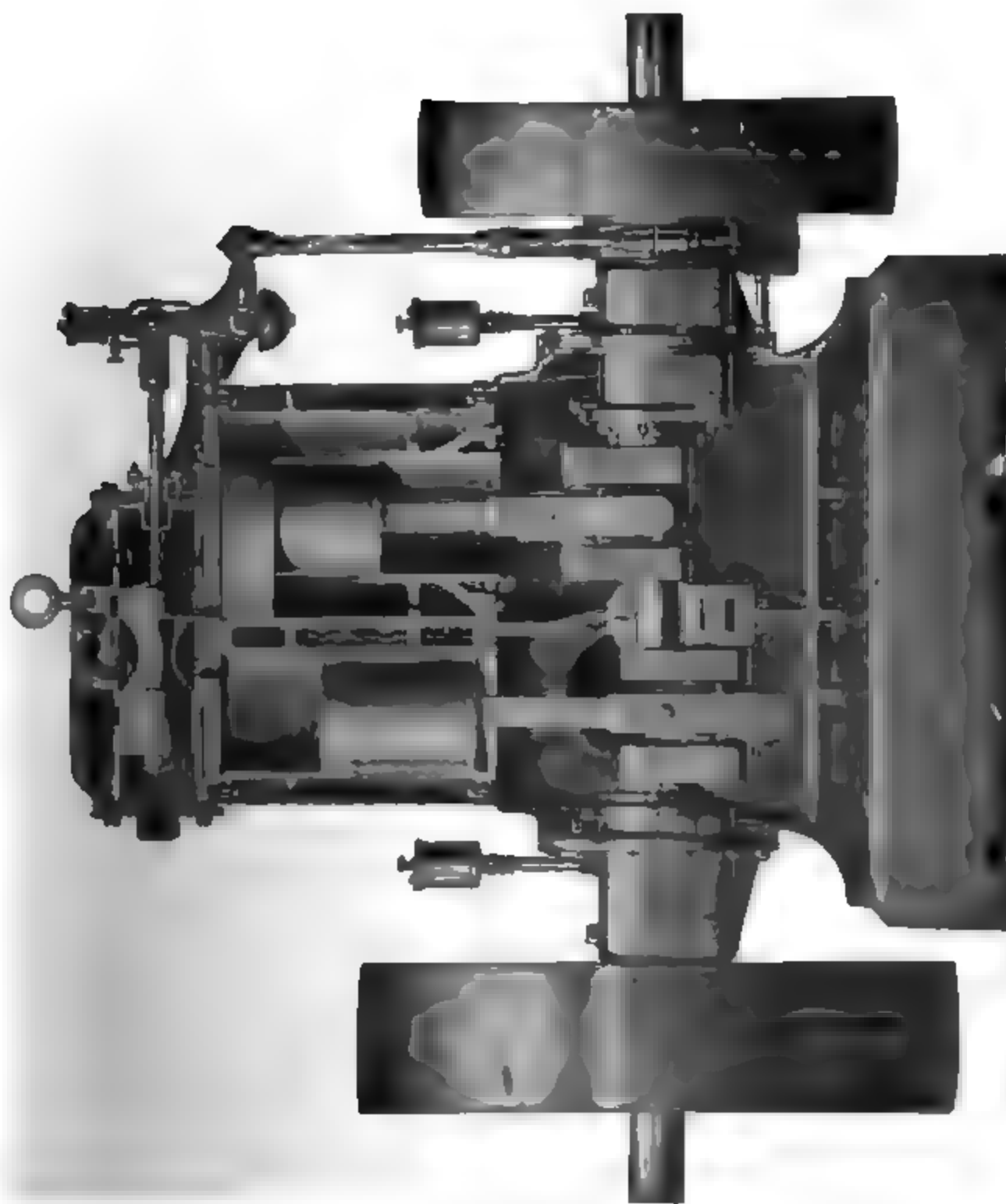


FIG. 208. — Section of Westinghouse Compound Engine.

arrangement. For this make of engine a tandem compound is chosen as representative, differing from the simple engine only in the arrangement of the cylinders. Under these, of course, a support is necessary, and it is placed beneath the connecting-piece which joins them. The features to be noted at this point are, the construction of the engine case, of sheet-metal, the steam-piping between the cylinders, the use of piston-valves (as also in Fig. 205), and the form of the frame for the generator.

(f) THE WESTINGHOUSE SINGLE-ACTING ENCLOSED ENGINE, illustrated in Figs. 207 and 208, and in Fig. 255, has a number of features in which it differs from the conventional type. Most important is the trunk piston, which also serves as cross-head, in a manner more clearly shown by Fig. 255. In the simple engine of this type there are two pistons of equal size; here the low-pressure cylinder is made with two diameters, in order that there shall not be variation in the volume of the enclosed space about the working piston. The trunk piston can, of course, receive steam-pressure at one end only, although engines have been built—in early marine practice—with a double-acting piston like that in the low-pressure cylinder of this engine. Here the annular space around the low-pressure trunk forms a cushion-chamber, in which a confined body of steam alternately expands and is compressed, with some effect upon the force-action in the machine.

This form of the engine-mechanism has the advantage that it shortens up the machine considerably; and with the trunk piston, as with the outside-packed plunger in a pump, leakage is bound to make itself immediately evident. The cranks are necessarily opposite, whether the engine is simple or compound; for with any other angle than  $180^\circ$ , there would be a time in the revolution when both pistons would be rising or both descending; and this in a single-acting engine, would be worse than a dead-center. In regard to work-performance, the two pistons of this engine are equivalent to the single piston of the double-acting engine.

Referring to § 40 (1), we see that this machine is practically self-balanced against direct shaking-force; and with the cylinders so close together there can be only a small twisting shake, so that counterweights on the cranks are not needed.



The crank-chamber is filled with water to a height such that the crank-pin is about half immersed when in its lowest position, and on this water floats a half-inch layer of thick black oil. When the engine is running, an emulsion of oil and water is continually churned and splashed about, so as to lubricate thoroughly the pins and the piston-slide.

(g) CHARACTERISTICS OF THE HIGH-SPEED TYPE.—With the preceding example in view, the cogency of the following statements can be fully appreciated:

The high-speed engine has a short stroke relative to the diameter of its cylinder, which determines that the whole machine shall be short and compact. The framework is usually a single casting, of rather complicated form; and the machine in general is very much of the “self-contained” type.

With the short stroke goes a high rotative speed, or a high frequency of stroke, in order that a good mean speed of the piston may be maintained. The latter, usually expressed in feet travelled per minute, is a primary factor in the piston-displacement, upon which depends the power developed by the engine. While in no sense a drawback when the engine is to be used for such work as driving shafting, this high speed of rotation adapts it especially to the direct driving of electric generators of moderate diameter. A discussion of engine-speed will be found in Art. (u).

All the engines thus far shown have slide-valves, of either the flat or the piston form; and these are driven and controlled by shaft-governors, which regulate the steam-distribution so as to accommodate the power of the engine to its load. In every case, the valve remains always under the control of the eccentric, or the valve-gear is positive: except as they act upon the governor so as to change the position of the center of the eccentric, the forces in the valve-gear do not affect the movement of the valve, unless they be strong enough to cause elastic change of shape or other disturbance in the parts of the mechanism.

To recapitulate, the high-speed engine is characterized by a short stroke and compact construction, by high rotative speed, and by a positive valve-gear, usually with automatic regulation.

The older and cruder type of engine with fixed eccentric and

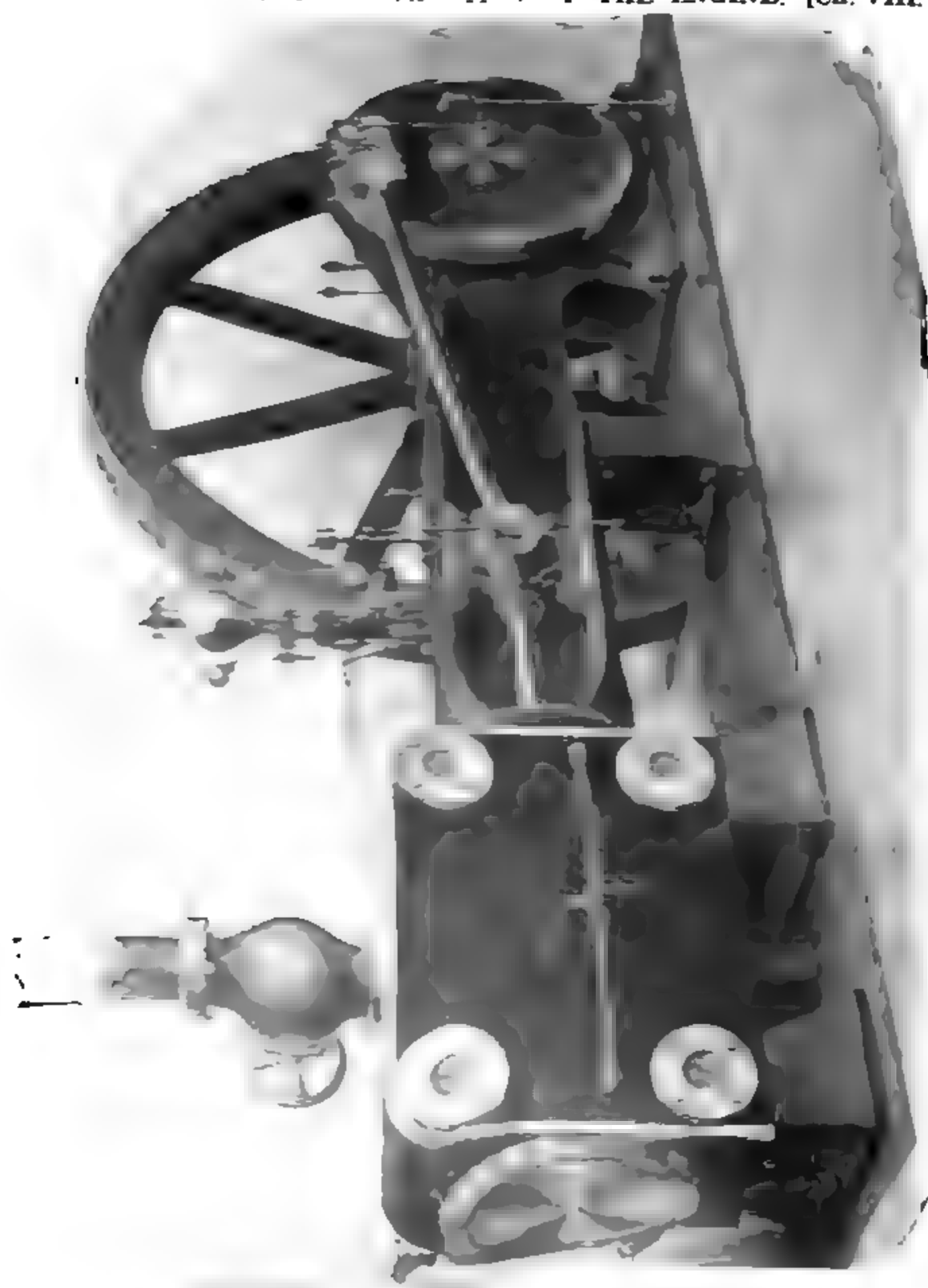


Fig. 269. Compound engine with standard cylinder head. Atlas Engine Works.

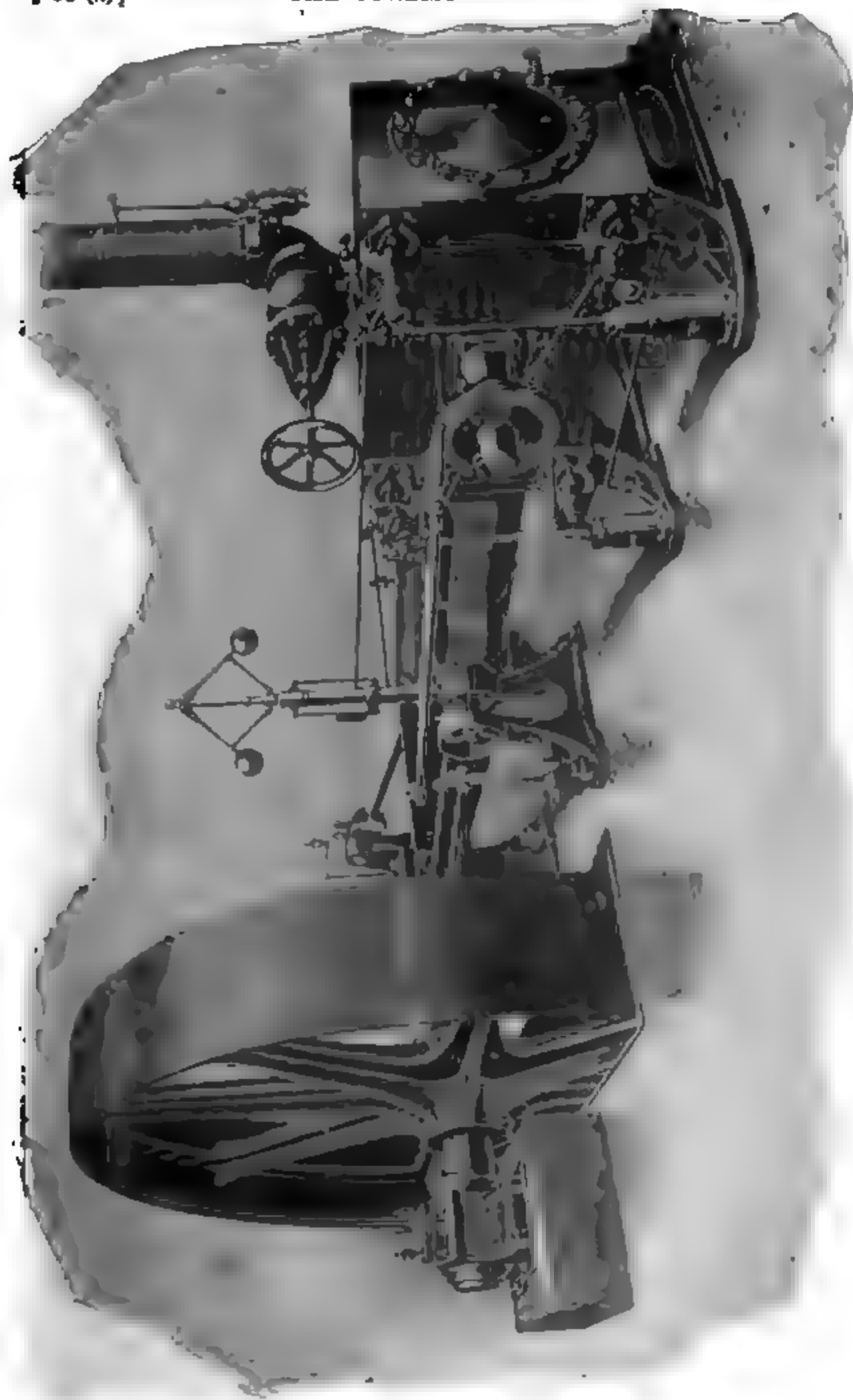


FIG. 210.—Fishkill-Corliss Engine with Girder Frame. Fishkill Landing Machine Company.



throttling governor would generally fall into the class under discussion; from the points of view of form and valve-gear, if not always of running speed.

To present the exception which every rule requires, it may be remarked that one of the pioneer designs of the fast-running engine, the Porter-Allen, has cylinder proportions similar to those found in engines of the Corliss type; that is, the stroke bears a high ratio to the diameter.

(h) SIMPLE CORLISS ENGINES.—In Figs. 209 to 213 is given a series of examples of Corliss engines, showing the important variations in form. As to the cylinder, of which sections will be found in Figs. 261, 268, etc., it has usually a rectangular external shape, when the lagging or covering is in place. It is strongly supported by the foundation, resting upon two feet at the ends as in Figs. 209 and 210, or upon a spreading base-plate as in Fig. 213, or with everything inside the sheathing as in Figs. 211 and 212. There are separate valves, of the oscillating plug type, at the four “corners” of the cylinder, those at the top for admission, the others for exhaust; with a rather complex-looking valve-gear at the back to operate them.

Of this valve-gear the “releasing” action is an essential feature. The steam-valves are under the control of the eccentric during only a small part of the revolution, while they are being opened; at a point determined by the governor, the valve is released or unhooked; and is then promptly closed by the vacuum-cylinder or dash-pot, remaining at rest until the time for a new admission.

These engines are run at moderate speeds—in the great majority of cases from 80 to 100 R.P.M.—and therefore have a long stroke, which lengthens out the whole machine. The girder-frame, shown in Figs. 209 and 210, has long been the standard form for Corliss engines: its general shape is clear from the illustrations, Fig. 210 showing the flanged stiffening-rib formed on the back of the casting. In this example, representing the older type of design, girder-frame and main-bearing pedestal are separate pieces, bolted together; in the heavier construction of Fig. 209, they are cast in one piece.

For heavy duty—that is, for higher speed and higher steam-pressure, with heavier construction—most builders change to the



Fig. 10. The engine of the locomotive, with the boiler.

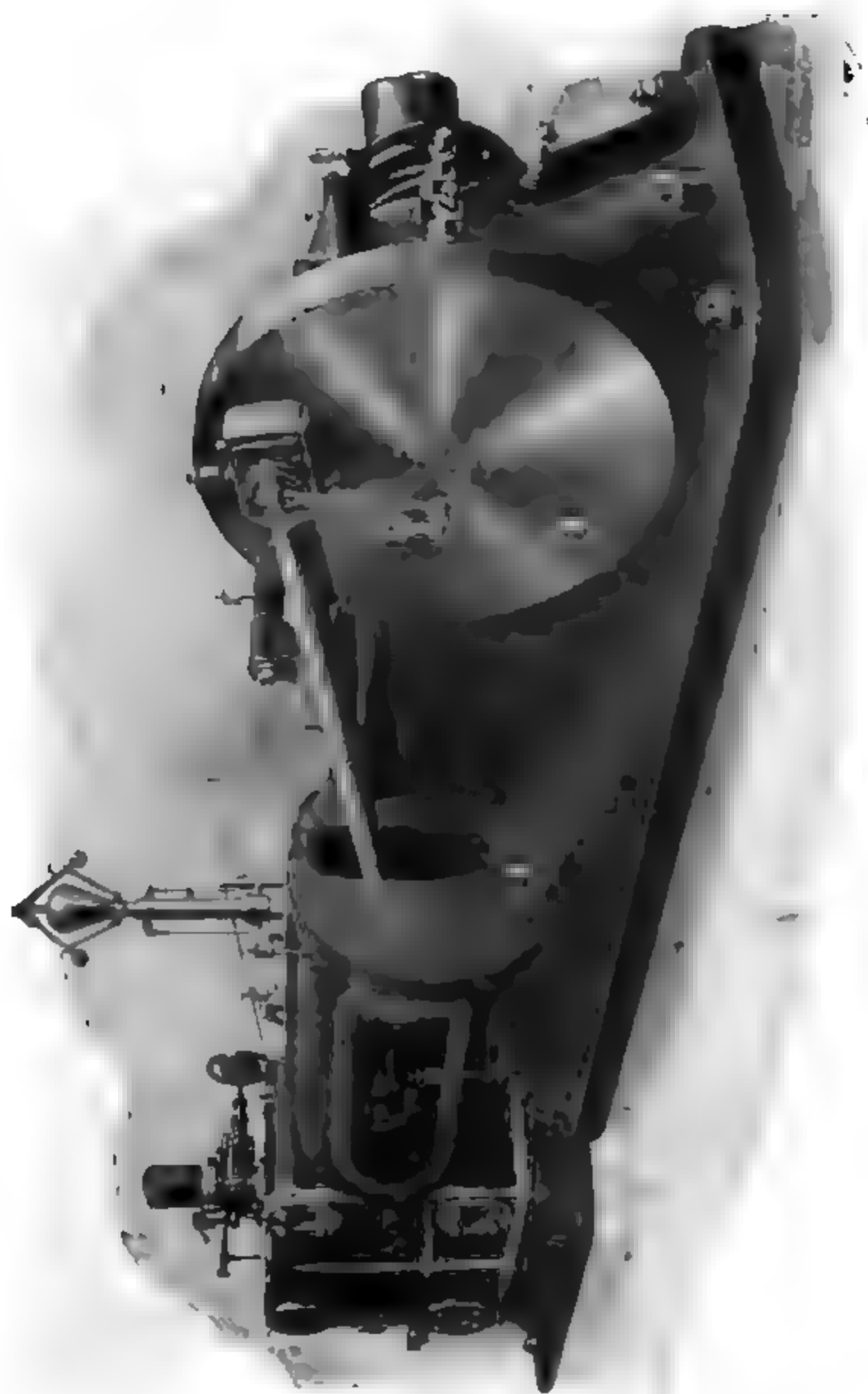


Fig. 213. - "Rolling-mill" Type of Corliss Engine. Murray Iron Works.



FIG. 211 Gram-Whitlock Engine in Electrical Service



form of frame shown in Figs. 211 and 212. The guides are cylindrical, the guide-barrel forming a separate piece; this is usually supported like a bridge, between the cylinder and the main frame, as in Fig. 211, but sometimes has its own base extending to the foundation, as in Fig. 212. In any case, the main frame is of the Tangye form, with broad base, surrounding the "crank-pit." It will be noted that the Cooper engine is provided with an oil-guard very similar to that on Fig. 203.

Fig. 213 is representative of the most recent heavy-duty designs, the frame being all in one piece, and having a spreading base to rest upon the foundation. With stiffening by judiciously placed internal ribs, this makes a very strong and rigid casting, and a frame which will not deflect under any force that can properly come upon it. This self-contained type of construction facilitates the securing and the maintaining of the perfect alignment which is essential to smooth and easy running.

(i) THE CORLISS TYPE.—The characteristics of this type of engine are, a long stroke, low speed of reciprocation, a drawn-out and open form, separate valves, and a releasing valve-gear. As already stated, engines of other designs also have these same features; but it seems fair to take the most prominent member of the class, the Corliss, as the type of all.

Next in importance is the Wheelock Engine, of which an example is given in Fig. 214. There are still four valves, but the two for each end of the cylinder are grouped together and placed at the bottom. This leads to a cylinder quite different in external appearance from that of a Corliss engine. The valve-gear too is radically different in form (though not in effect), consisting of a side-shaft along the back of the engine, which turns with the main shaft and carries an eccentric for each valve. The form and arrangement of the valves can be seen in Fig. 266; they are of the gridiron type, with flat face and a short reciprocating movement. A description of the valve-gear will be found in the next chapter.

Very common in Europe, though rare in American practice, is the engine with lift- or poppet-valves—or, as they are called when the valve-gear is of the releasing type, drop-valves. The cylinder

of one of these engines is quite fully illustrated in Fig. 267; valves are located as in a Corliss engine, and the primary part of valve-gear is like that of the Wheelock engine, in that there is a light shaft along the back of the engine, driven from the crank shaft by mitre-gears, and carrying an eccentric for each valve. The valve-gear proper, at the cylinder, is illustrated in § 59.

(j) COMPOSITE TYPES.—This title seems best to cover the class of engines represented by Fig. 215. This has Corliss valves and the peculiar valve-movement of the Corliss gear, but the rele-

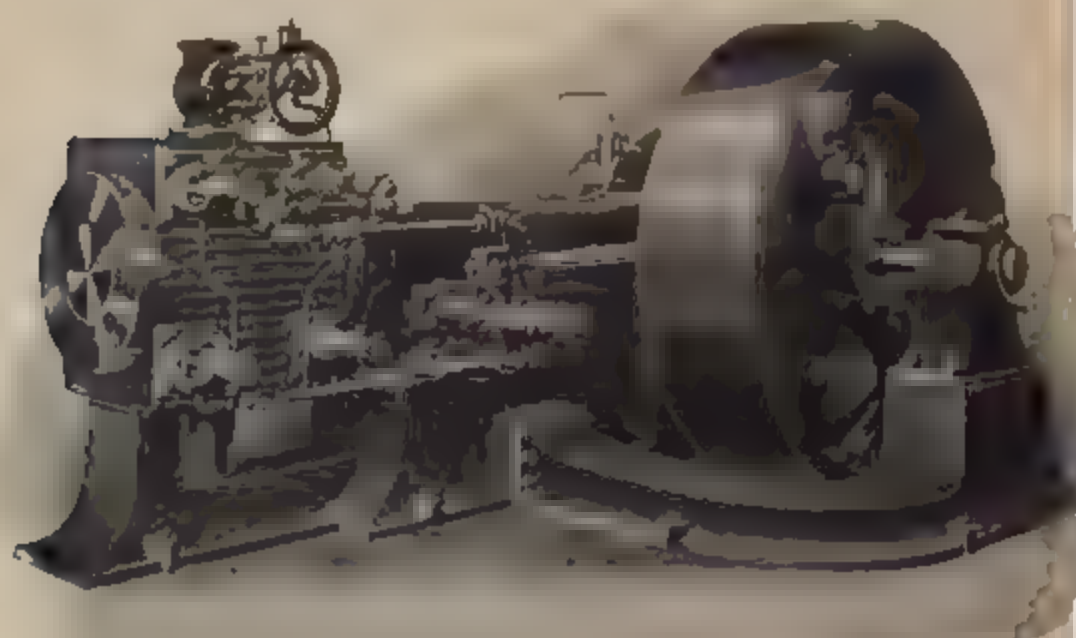


FIG. 215    Fleming Four valve Engine.    Harrisburg F. & M. Co.

feature is omitted: instead, the eccentric-center is shifted to the crank shaft-governor, as in the ordinary high-speed type—and these engines are intended to run at high speeds. Distinctions thus made upon the form and action of the valve-gear can be much better understood after a perusal of the next chapter, but it will not be amiss to remark here that an essential feature of the Corliss valve-gear is a marked distortion of the movement of the valve from simple harmonic motion that is produced by the eccentric; this effect is now introduced into a number of positive valve-gear—the engines shown in Figs. 216 and 218 being further examples.



FIG. 216.—Cross-compound Engine, 1800 H.P. McIntosh, Seymour, & Co.

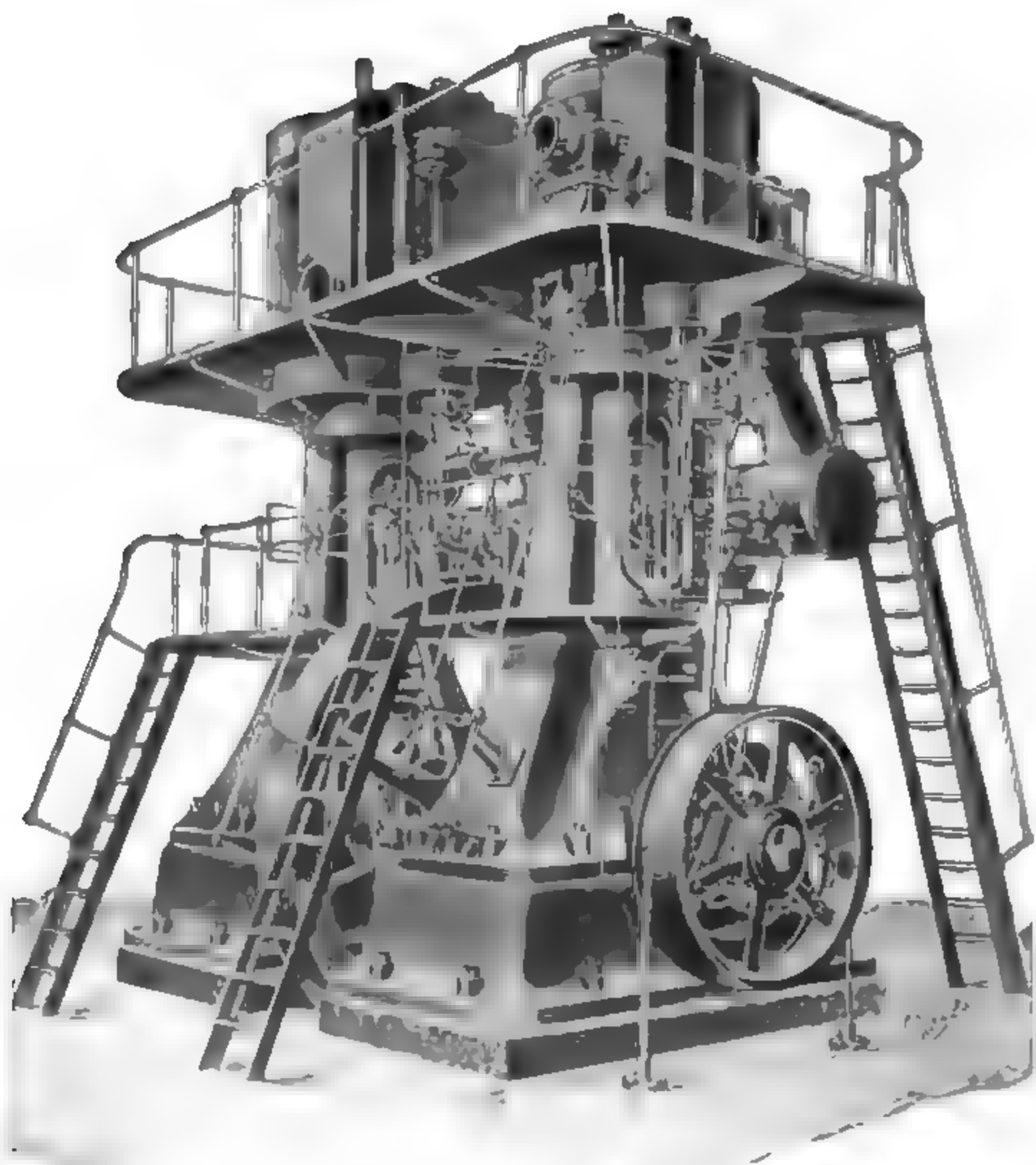


FIG 217. - Vertical Cross-compound Buckeye Engine.

(k) LARGE STATIONARY ENGINES, especially for power-house service, are well represented by Figs. 216 to 219. The first example shows the horizontal, cross-compound type, which has its cylinders side by side and the cranks at right-angles. Structurally, an engine of the type here shown usually consists of two self-contained machines, resting on the same foundation, and joined mechanically by the shaft. These elements may form the two parts of a compound engine, as in this case: or each may be a complete tandem-compound engine, making four cylinders in all: or, with more complex steam connections and a proper gradation in diameter of cylinders, the engine may be triple- or quadruple-expansion. To get four cylinders in a triple engine, two equal low-pressure cylinders are used, one on each side. Small horizontal, high-speed, cross-compound engines have usually the compact construction typified by Fig. 202.

In Fig. 216, the engine has an inside crank on the high-pressure side, with a coupling on the outer end of the shaft, for connection to a line-shaft or a generator: but usually, in direct-connected units, the generator is put on the middle part of the shaft, beside the fly-wheel, as in the vertical engine shown at Fig. 218. This engine has a positive valve-gear of special form—see § 58—with the cut-off controlled by a shaft-governor. The majority of large power-plant engines are, however, of the Corliss type.

High-grade vertical engines are shown in Figs. 217 and 218. As to the form and arrangement of the cylinders, nothing need be said, except to call attention to the steam-connections: in both cases, the receiver is a long cylinder back of the frame, large enough to contain a re-heating steam-coil. The frames are simple housings of graceful outline, made in two or more pieces, and completely enclosing the working parts. The bed-plates are necessarily independent of each other, to leave room for the wheel, and usually for the generator, between them. The idea of a combination of mechanically complete elements, united into one structure chiefly by the foundation, is strongly exemplified in the general arrangement of both these engines—in which they differ markedly from the compact and self-united marine engine, as illustrated in Figs. 232 to 235, which has not a heavy and rigid foundation whereon to rest.

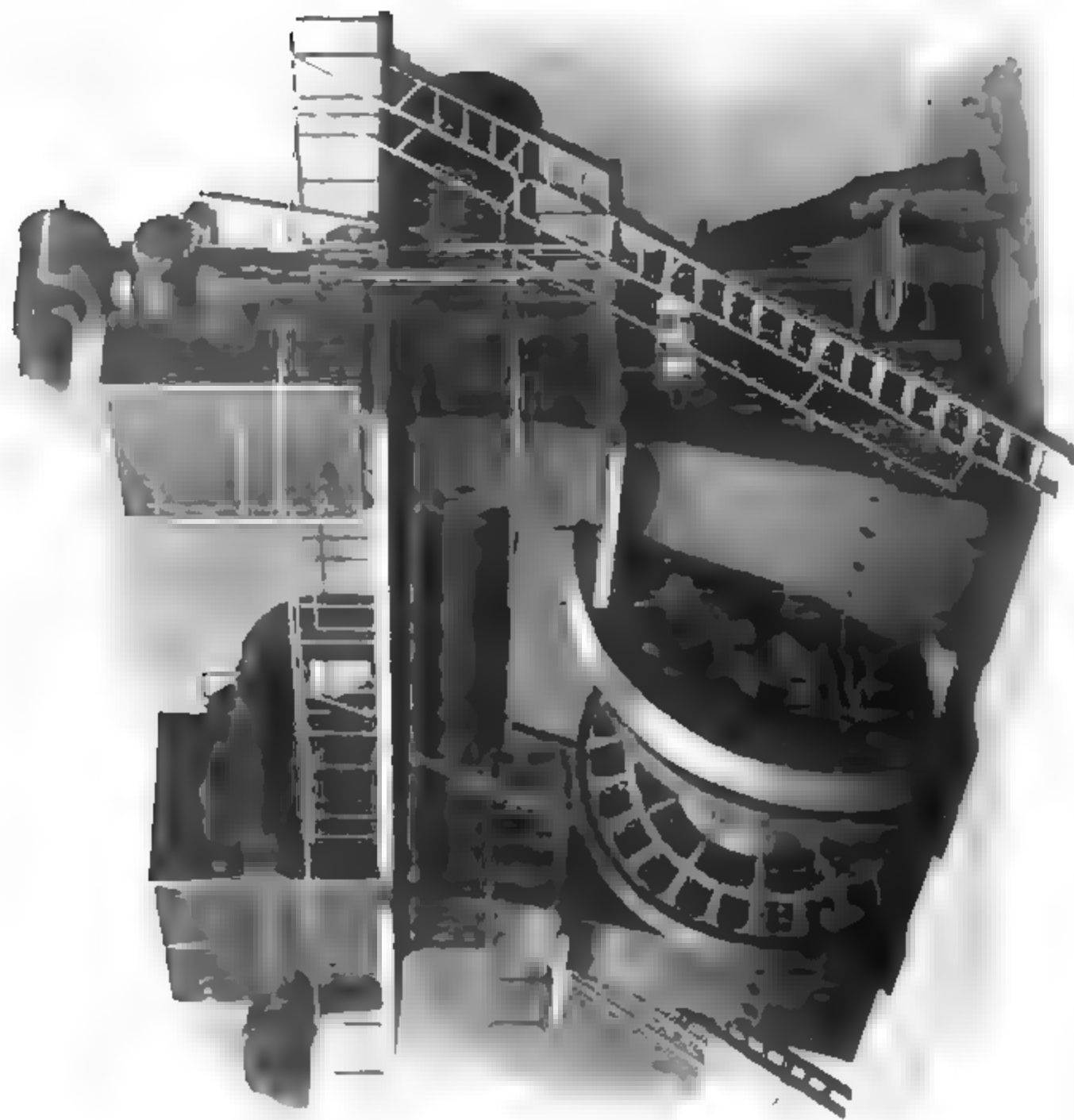


FIG. 218. - McIntosh & Seymour Compound Engine. Cylinders 40" and 83" by 54"; R.P.M. 100; 3840 H.P.

Fig. 217 is shown as an engine alone, without its load. It could drive two generators, one coupled to each end of the shaft; or, by separating the halves, room can be made for a larger generator in the middle. It must be understood that when engines get above a certain limit of size, say from 600 to 800 H.P., there is not the same standardization in designs that is usual in the smaller sizes:

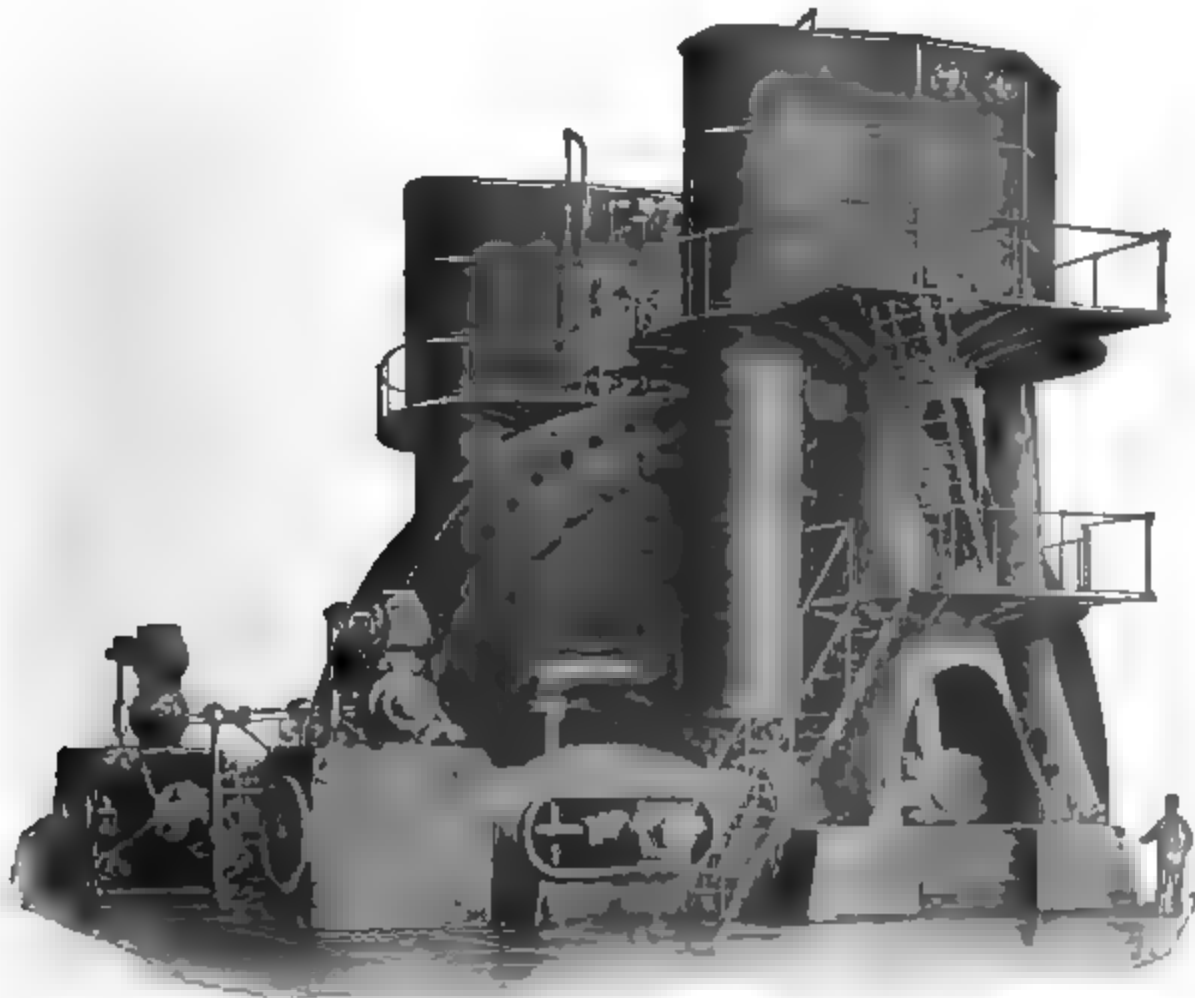


FIG 219.—Reynolds Horizontal-Vertical Duplex Compound Engine, "Manhattan Type." Allis-Chalmers Company. Cylinders 44" and 83" by 60"; R P.M. 75; 8000 H.P.

it is necessary rather to modify and adapt general designs to particular conditions, though with the least possible variation in the detail of the parts.

A special design for large generator units, which is coming into increasing prominence, is shown in Fig. 219. It is a duplex compound engine, each half the combination of a horizontal high-

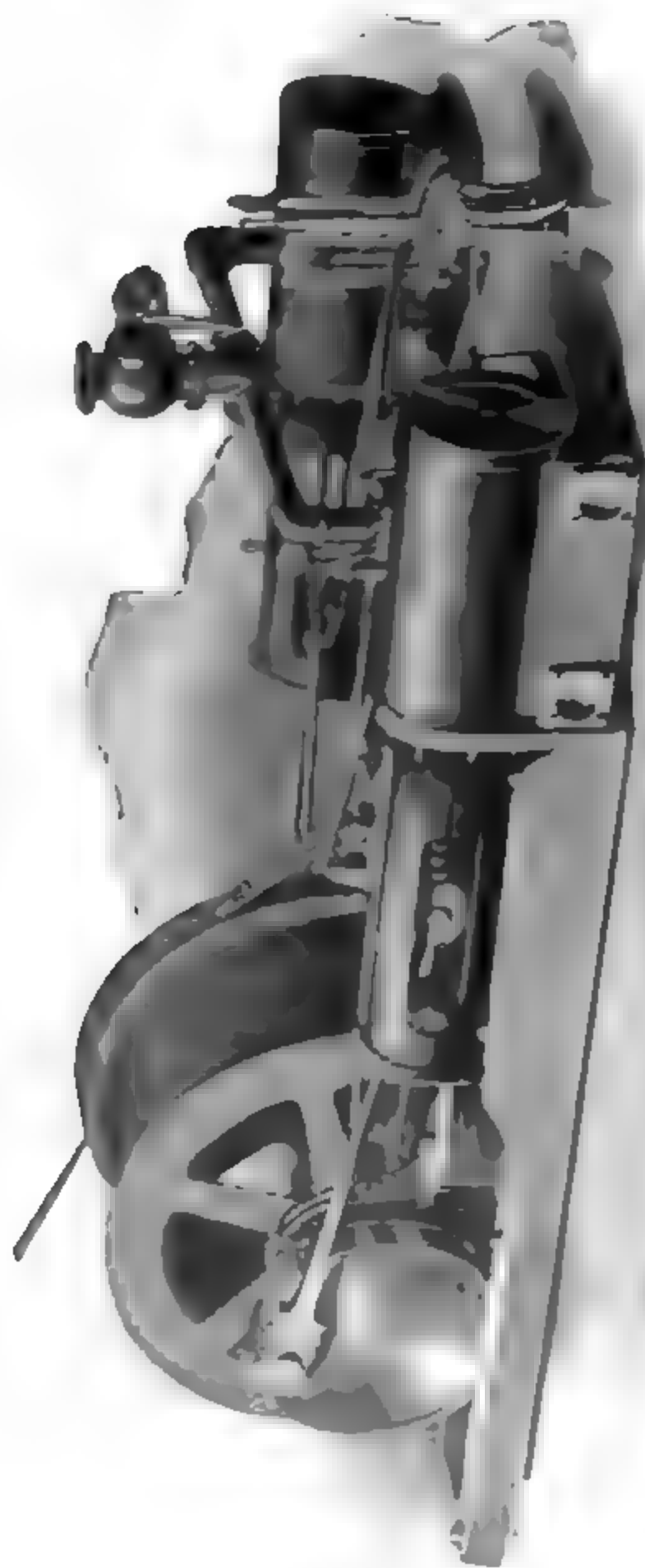


FIG. 220. Single-drum hoisting engine. Allis-Chalmers Company.



pressure engine with a vertical low-pressure, with the two connecting-rods working on one crank: by setting the two cranks at an angle of  $135^\circ$ , a very uniform turning-moment is secured; and the rotor of the generator, here the field, is the only fly-wheel required. The horizontal frame is like that in Fig. 212; the vertical housing is best described, perhaps, by naming it the "tree-trunk" type. Details of the low-pressure cylinder, with the valves in the heads, are given in Fig. 268. The re-heating receiver, above the horizontal engine, and the large exhaust-pipe beyond the housing, fill out the lines of the vertical engine.

(l) HOISTING AND ROLLING-MILL ENGINES.—Coming now to the arrangements in which the engine is directly connected to the working machine, we have first the two cases of a rotary load exemplified in Figs. 220 and 221, both engines being of the duplex-simple, reversing type. Small hoisting-engines are usually of the semi-portable form, mounted on a bed-plate with a vertical boiler. Fig. 220 shows a very simple form of the large mine-hoist, the designing of which is always a special problem for the engineer; the complete apparatus for this service showing a wide variety in form and arrangement. In some cases, with deep shafts and almost continuous operation of the hoist, and where fuel is expensive, compound engines are used: but where there are frequent stops and starts, with periods of rest, the economy of the compound has no chance to develop itself and overbalance the mechanical disadvantage of sluggishness in responding to the throttle-valve. The engine here illustrated is called direct-acting because the drum is placed directly on the engine-shaft, without gearing. The valves are of modified Corliss form, positively driven by a link-motion valve-gear. The engine is hand-controlled: one of the levers shown is for reversing, the other for working the brake; and the throttle-valve is of the quick-opening type, operated by a lever.

The rolling-mill engine in Fig. 221 is of the kind that reverses for each pass of the metal between the rolls, or that drives a two-high roll-train—as distinguished from the continually running engine on a three-high train. In either mode of running, the engine is subjected to exceedingly severe force-actions, and must be made very strong and heavy. The reversing-engine has no fly-wheel,

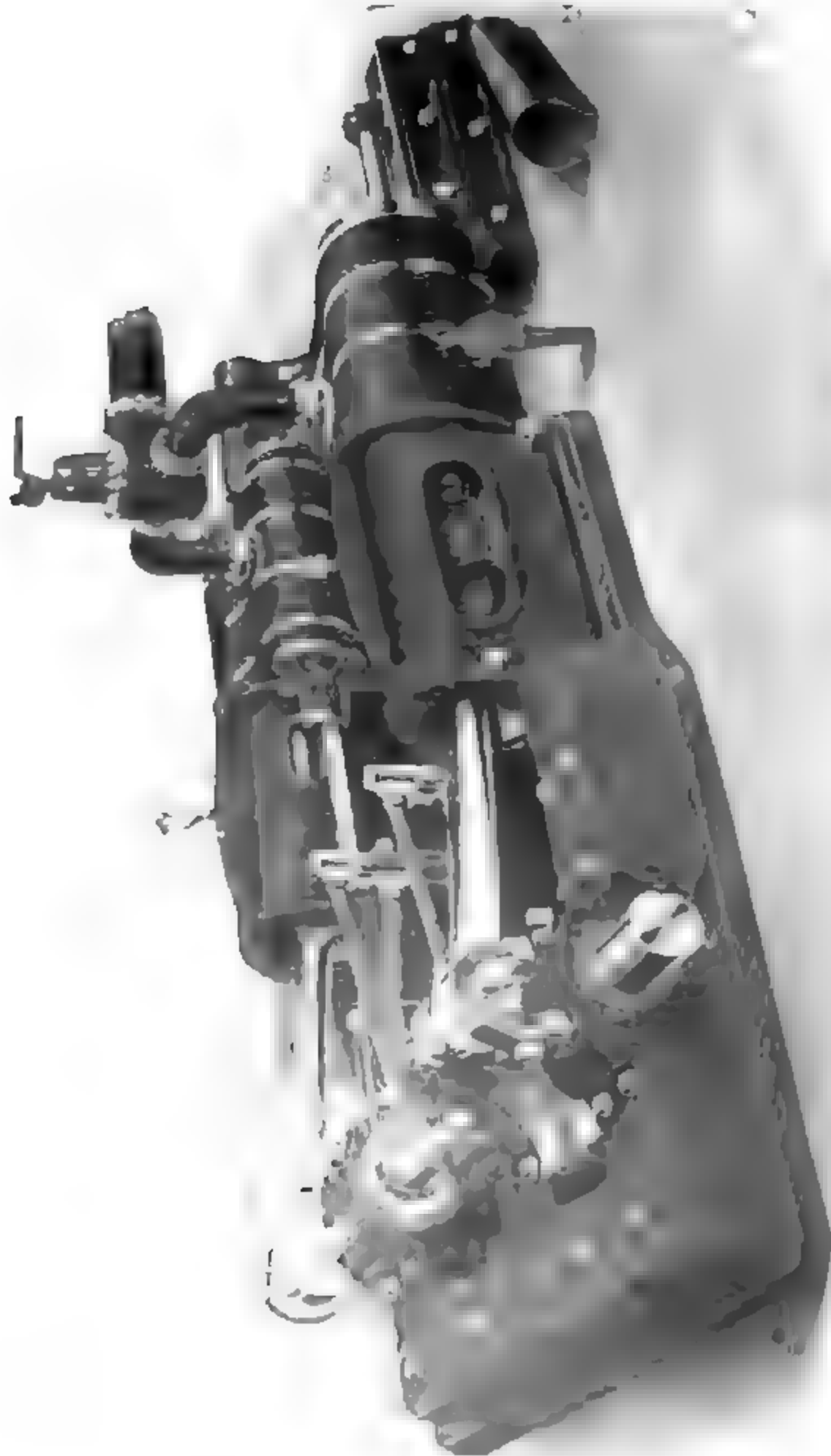


FIG 221. Reversing Rolling-mill Engine. The William Tod Company. Two Single-expansion Cylinders 54" x 66".

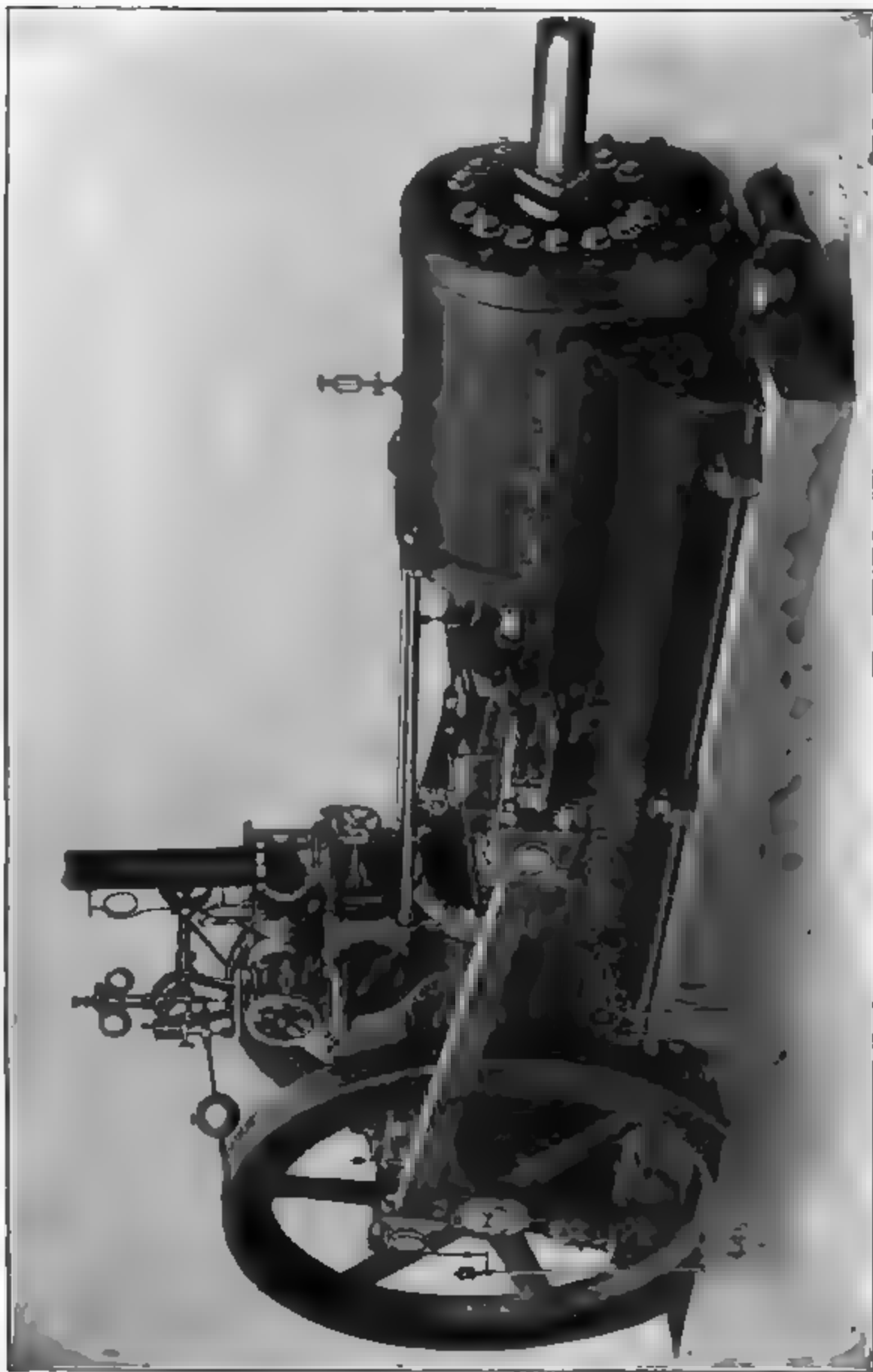


FIG. 223.—Ingersoll-Sergeant "Straight Line" Air-compressor.

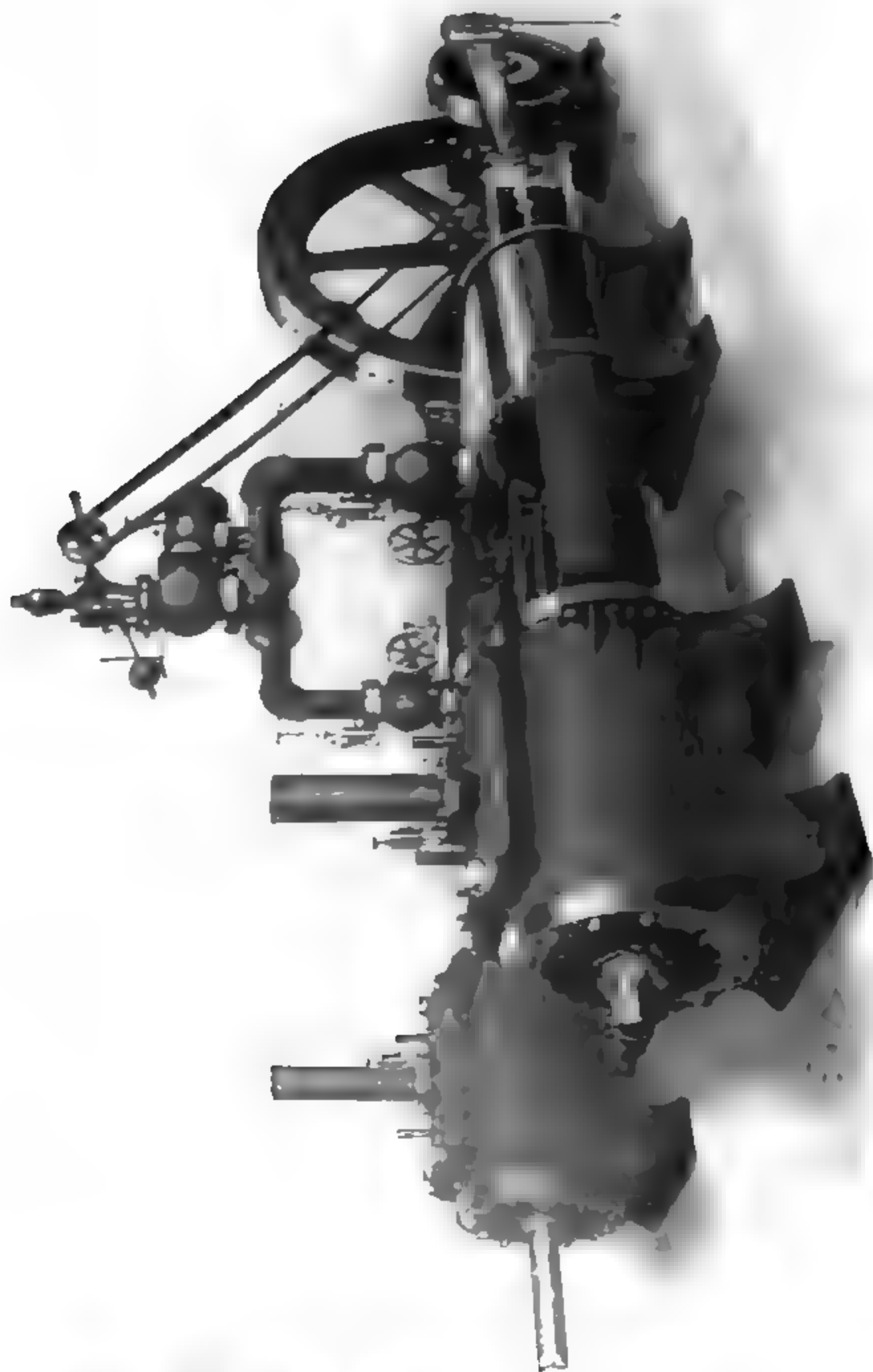


FIG. 223. Duplex Air-compressor, Extended Type.

necessarily, and is always of the duplex, single-expansion type. It is hand-controlled, with a special apparatus for quickly moving the heavy reversing-gear, shown in Chapter X. It will be noted that the bed-plate of this engine has a different shape at the two sides, since one element is of the center-crank, the other of the side-crank, form. The largest piece of the frame, on the center-crank side, weighs about 84 tons, and the whole engine about 500 tons.

(*m*) AIR-COMPRESSORS AND BLOWING-ENGINES.—The distinction between these two classes of machines is based on service and air-pressure produced, with resulting great differences in form. “Compressed air” is used chiefly for driving rock-drills and the other machines employed in mining and quarrying, and for small tools and portable machines in workshops. For all this line of service, the pressure is usually from 60 to 100 lbs. per square inch; but there are other cases where very high pressures are required, running up to several thousands of pounds per square inch. The function of the air-compressor is, then, to deliver a relatively small volume of air at a high discharge-pressure. The blowing-engine, on the other hand, must deliver very large volumes of air, but with a low ratio of compression; in blast-furnace work the pressure is usually somewhere between 15 and 30 lbs. per square inch (above atmosphere, of course), and the compression-ratio is from 2 to 3.

A single air-compressor, the simpler and more compact form of the machine, is shown in Fig. 222. The steam-cylinder is between the cross-head and the shaft, so that the engine proper has what is called the “return connecting-rod” arrangement, which makes it very short. The valve-chest is on top of the cylinder; and the running of the machine is controlled by a compound governor, which shuts off steam at a certain speed and also when a certain air-pressure is reached. The compressor-cylinder is simple in external form. Air is drawn in through the hollow extended piston-rod, and the inlet-valves are in the piston: the discharge-valves are under the caps in the cylinder-head. Always, the cylinder of a compressor is surrounded by a water-jacket, to prevent it from becoming overheated, and there must be a constant circulation of cooling-water.

A duplex compressor of very simple outline is illustrated in Fig.

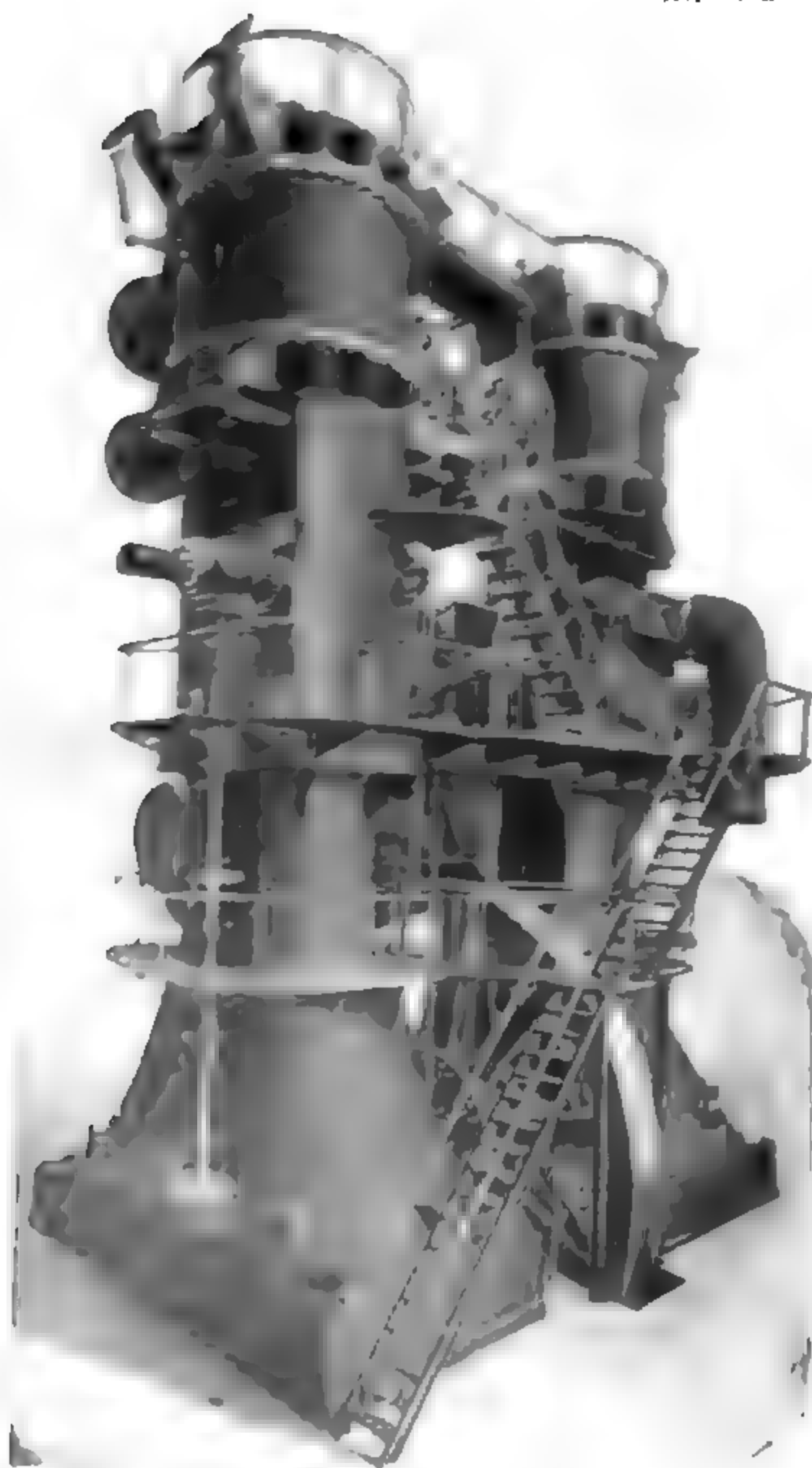


FIG. 334 Hot blowing engine at Ohio Works of Carnegie Steel Co., Youngstown, Ohio.

223. Here both steam-cylinders and air-cylinders are duplex, as well as the general form of the machine; but frequently the steam-engine is compound and the air-compressor works with two stages, just the reverse of those of the compound engine. There is then a cooling receiver, or inter-cooler, between the cylinders, with a coil or group of pipes through which cold water flows.

For large installations, where economy becomes of greater absolute importance, the engine is given the Corliss valve-gear: and often the valves of the compressor are moved mechanically, instead of by the air itself as in the examples shown.

(n) BLOWING-ENGINES are almost always vertical. For moderate capacities, the same cylinder-arrangement as in the air-compressor at Fig. 222 is usual, with the shaft at the bottom and the air-cylinder at the top: this is called the long cross-head type. Another form, the steeple blowing-engine, analogous to Fig. 223 in general arrangement, is shown in Fig. 224. Up to the air-cylinders it is just a big cross-compound Corliss engine. The particulars of this engine are as follows: steam-cylinders 54" and 102" in diameter, twin blowing-cylinders 108" in diameter, stroke 60"; rated steam-pressure, 150 lbs. by gage; air-pressure 18 to 30 lbs.; speed, controlled by an adjustable fly-ball governor, 25 to 50 R.P.M. A more recent construction of the same design has the engine cylinders increased to 58" and 110" in diameter, the other leading dimensions remaining unchanged. The engine can compress 60,000 cu. ft. of free air per minute to 25 lbs. gage-pressure. Its weight is about 1,300,000 lbs.

These large, high-pressure blowing-engines have metallic valves on the air-cylinders, mechanically operated; but the study of these is outside of the province of this work. The cylinders are not water-jacketed, so that the compression raises the air to quite a high temperature—high enough to destroy leather valves such as were used with lower pressures.

In a third type, the compressor element takes the form of a complete engine, the air-piston being driven from the crank-shaft through the usual mechanism of connecting-rod and cross-head: then steam-engine and air-pump are combined into a machine which looks much like any duplex vertical engine. The advantage

of this arrangement is, that by choosing the proper angle between the cranks, the two curves of turning-force—of the steam and the air—can be made to agree closely throughout the revolution so that the machine will run smoothly, even at low speeds. If the maximum driving-force on the engine-crank comes much earlier in the half-revolution than does the maximum resistance on the compressor-crank, the former is placed somewhere near either 90 degrees behind or 135 degrees ahead of the latter. Sometimes the steam-engine is horizontal, the blower vertical: and this arrangement is usual in machines for compressing ammonia or some refrigerating medium.

(c) STEAM-PUMPS.—To the type of machine represented by Figs. 225 and 226, with only the one main moving piece consisting of a piston and their rod, the name steam-pump is given. There is no mechanism to define the stroke of the piston, determining both its length and the manner of movement: so that we have here a case of free or purely force-controlled motion, as distinguished from the mechanically constrained motion of the engine with shaft and fly-wheel. The most interesting feature of these direct-acting steam-pumps is their valve-action. The main mechanism is elementary in both form and working: but the valve-gear, with its details of so controlling the steam-distribution that the stroke shall be of a certain length, is worthy of study: if the stroke is too short, there will be a waste of steam through excessive clearance-volume together with loss of pumping capacity; if it is too long, the piston will strike the cylinder-heads. A description of the details and working of several typical valve-arrangements will be given in § 60.

There are two typical forms of the steam-pump, the single and the duplex. Fig. 225 shows a small single pump, of the type frequently used for boiler-feeding. It would be possible to rig a pump through which the piston, when near the end of its stroke, would move the valve into the position proper for the next stroke: but any such direct-driving valve-gear is likely to act at low speed. Instead, the idea is universally applied of having the main piston move a small secondary valve, which controls the supply of steam to a secondary piston that moves the



valve. In the form of this device there is a very considerable variety, but the principle is the same all through.

In Fig. 226 is illustrated an example of the duplex pump, more complicated in that each steam element is a tandem-compound engine. The valve-gear as here shown is universally in use on pumps of moderate size, each piston moving the valve for the other side.

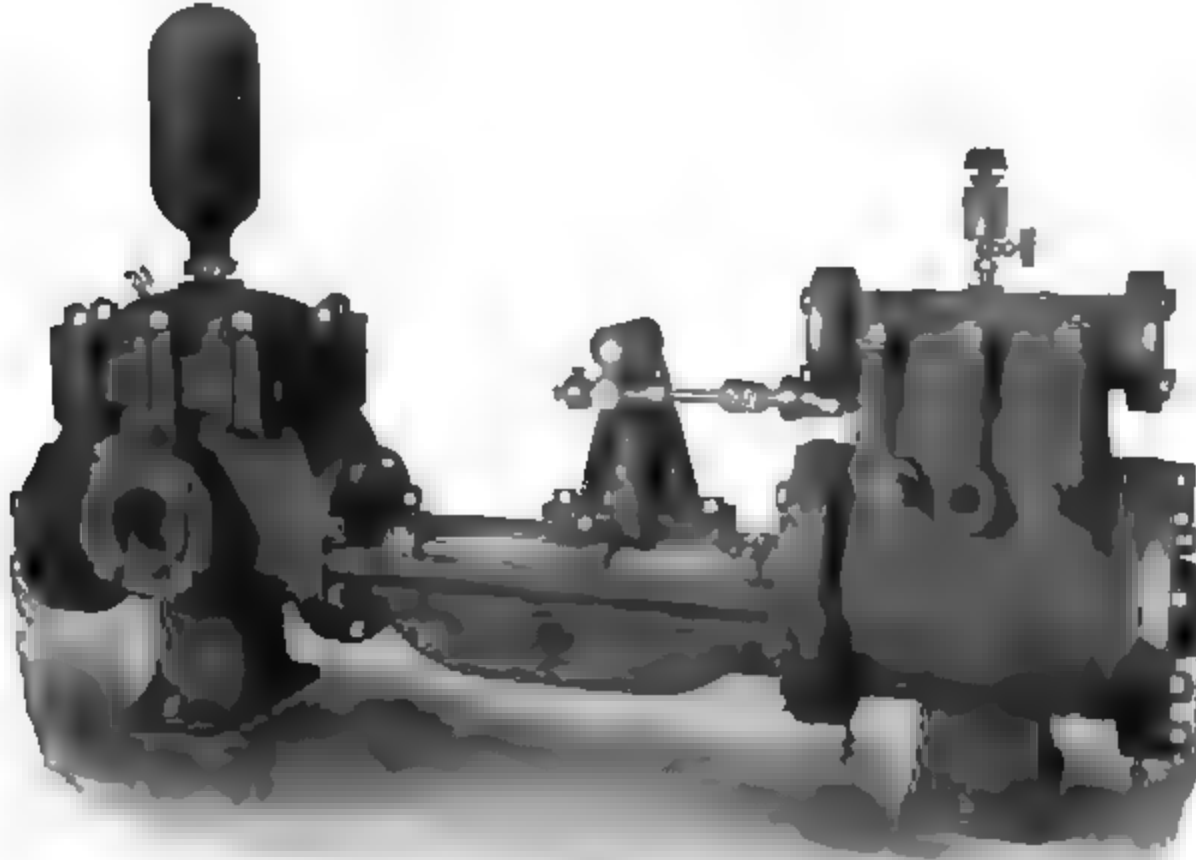


FIG. 225.—Single Boiler-feed Steam-pump.

The steam-driven water-pump works against a practically constant resistance in the water-cylinder, and secures an appropriately constant driving force by taking steam for its whole stroke. This makes it, necessarily, a very inefficient steam-engine. But when air is pumped, whether in the air-pump for a condenser or in the small air-compressor used to supply the air-brake system on a railroad train, there must be a still greater departure from the normal steam-cycle: because it is now necessary to have an effec-

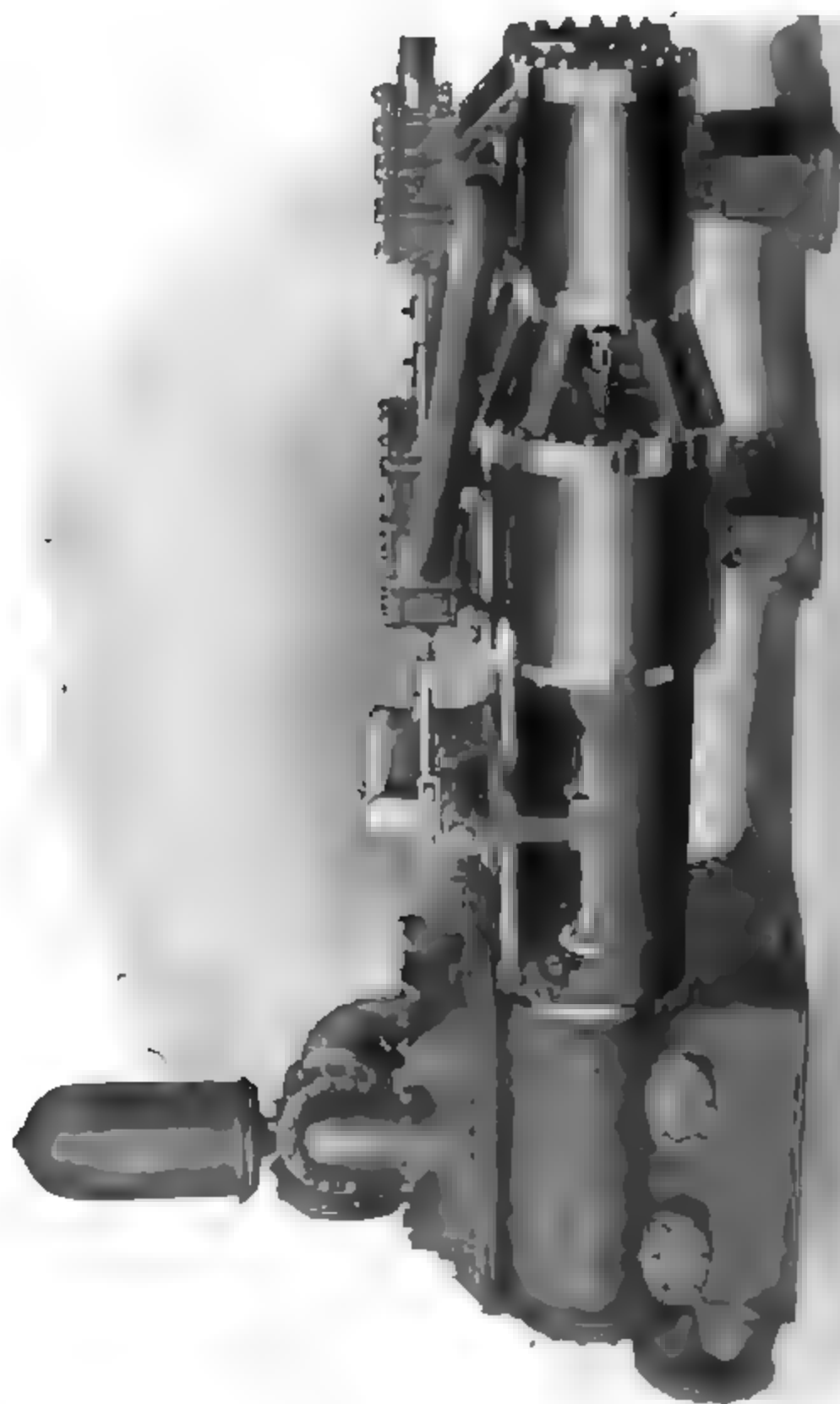


FIG 226. Duplex Compound Steam-pump.

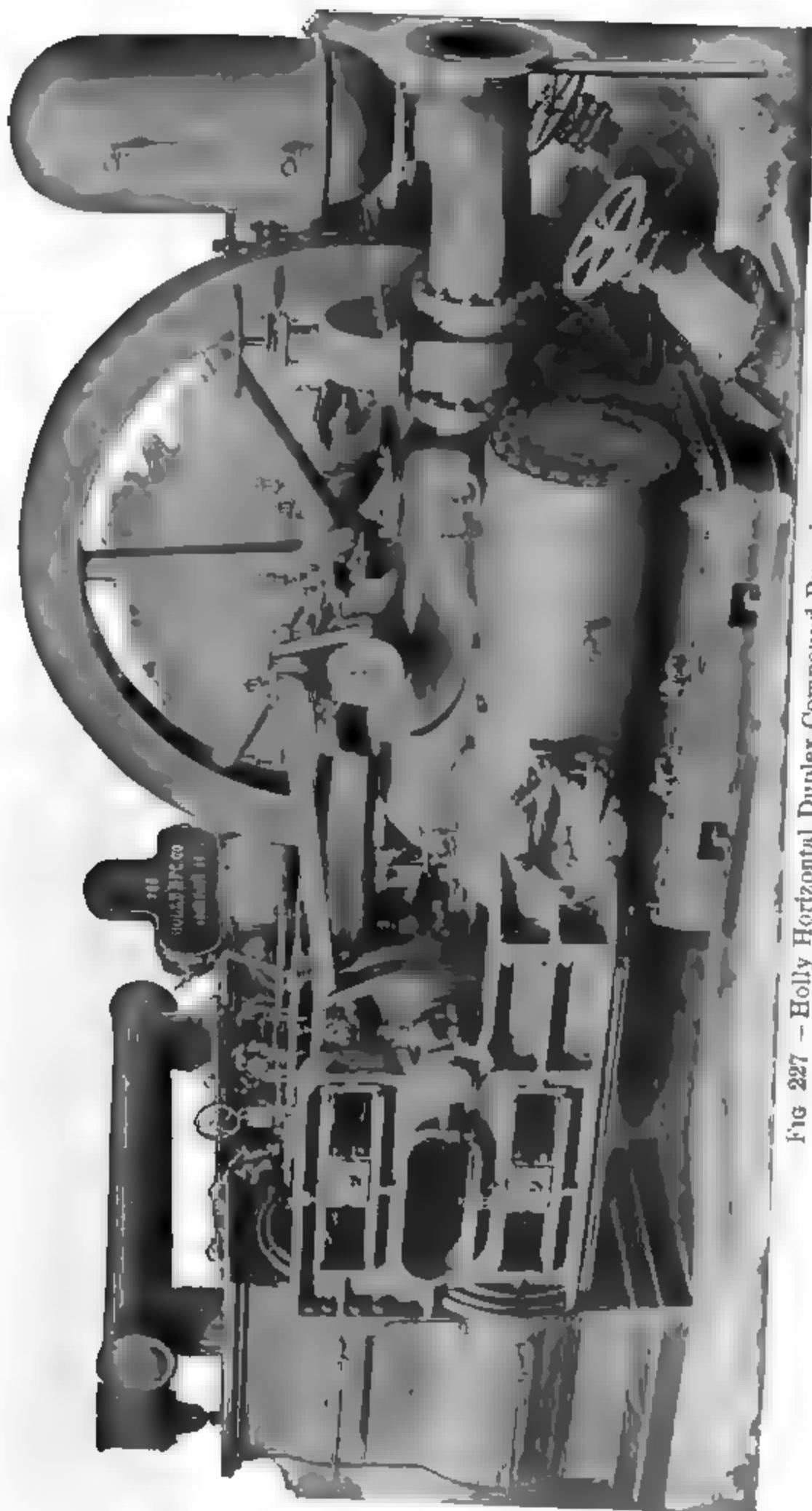


Fig 227 - Holly Horizontal Duplex Compound Pumping engine.



FIG 228. -- Worthington Duplex Triple-expansion Pumping engine.

tive driving-force which will increase toward the end of the stroke, and be greatest during the short period when the compressed air is being forced out against the discharge-pressure. Both single and duplex pumps are used for condenser air-pumps, but the standard air-brake pump is of the single type.

(p) PUMPING-ENGINES.—Different forms of the large water-works pumping-engine are shown in Figs. 227 to 230. The Holly engine is of the duplex quarter-crank form, the steam end on each side being a complete direct-expansion compound engine. The low-pressure piston-rod extends to the pump-cylinder, and the high-pressure cross-head is coupled in through a heavy vertical rocking-lever, as is further shown in skeleton outline on Fig. 236. The valve-gear is of the Corliss type, with some special features; and the cut-off, variable on the high-pressure cylinder, can be adjusted by hand, or placed under automatic control.

Fig. 228 shows a large pumping-engine of the indeterminate-stroke type, developed from the steam-pump. The three cylinders are in line, but while the high and intermediate pistons are on one rod, the largest piston has two rods, which extend forward on both sides of the smaller cylinders, and engage the projecting ends of the cross-head. The valves and gear are of the general Corliss form; but the releasing feature is omitted: the main wrist-plates (the circular disks at the middle of each cylinder) are oscillated by the cross-head on the other side; but the small auxiliary plates which control the steam-valves receive an additional movement from the cross-head on their own side. This combination gives the valve a complicated motion; the difficulties in the way of an exact determination of this motion being increased by the fact that there is no closely definite time-relation between the strokes of the two cross-heads.

The function of the fly-wheel in a pump or air-compressor is to equalize or average up the driving-forces and the resistance, throughout the revolution. If the steam is cut off rather early, the effective steam-pressure will be high at the beginning of each stroke, and will diminish as the piston advances. To store up the excess steam-work in the first part of each stroke, and give it back to make up the deficiency toward the end, a special equalizing de-



FIG. 289.- Snow Horizontal Triple-expansion Pumping-engine

vice is used instead of a fly-wheel on the machine shown in Fig. 228; this is drawn in outline on Fig. 236, where the working is explained.

Fig. 229 shows a fine horizontal triple-expansion pumping-engine. It has three cranks at angles of 120 degrees, which arrangement not only produces a smooth-running engine, but also causes the water to be delivered with very little pulsation, or variation in pressure. The manner in which the driving-force is transmitted from engine to pump, by parallel rods displaced diagonally from the axis, so as to clear the connecting-rod and crank, is clear from the photograph. This engine has the regular Corliss valve-gear, with a fly-ball governor controlling the cut-off in all three cylinders: the special wide-range cut-off gear, often used, will be described in the next chapter. Above the cylinders are placed the two re-heating receivers.

The final example, Fig. 230, shows a type of machine very frequently used in plants of large capacity. One advantage of the vertical arrangement is, that if the pumping-station is located on a river liable to great changes in height, the pump-cylinders can be placed low enough for effective suction without danger of the engine being drowned by every flood. There is, of course, always the point in favor of a vertical engine that the weights of the moving parts are less likely to cause uneven wear of the sliding surfaces: and this matter is of more importance in a pumping-engine than in lighter and quicker-running machines.

In this pump, the water-cylinders—almost concealed behind the large air- and valve-chambers—are of the single-acting type; water is drawn in beneath the plungers as they rise and expelled as they descend. Besides the general arrangement here shown, another plan is to bring the main foundations up to the base of the engine proper, and then suspend the pumps. In this figure, the first gallery, at the engine-shaft, marks the level of the main floor of the pump-house. The valve-gear is of a modified Corliss form, and is driven, not by the main shaft, but by an auxiliary shaft on the front of the engine; and this is driven by two parallel connecting-rods, at the two ends, with the cranks at right-angles.

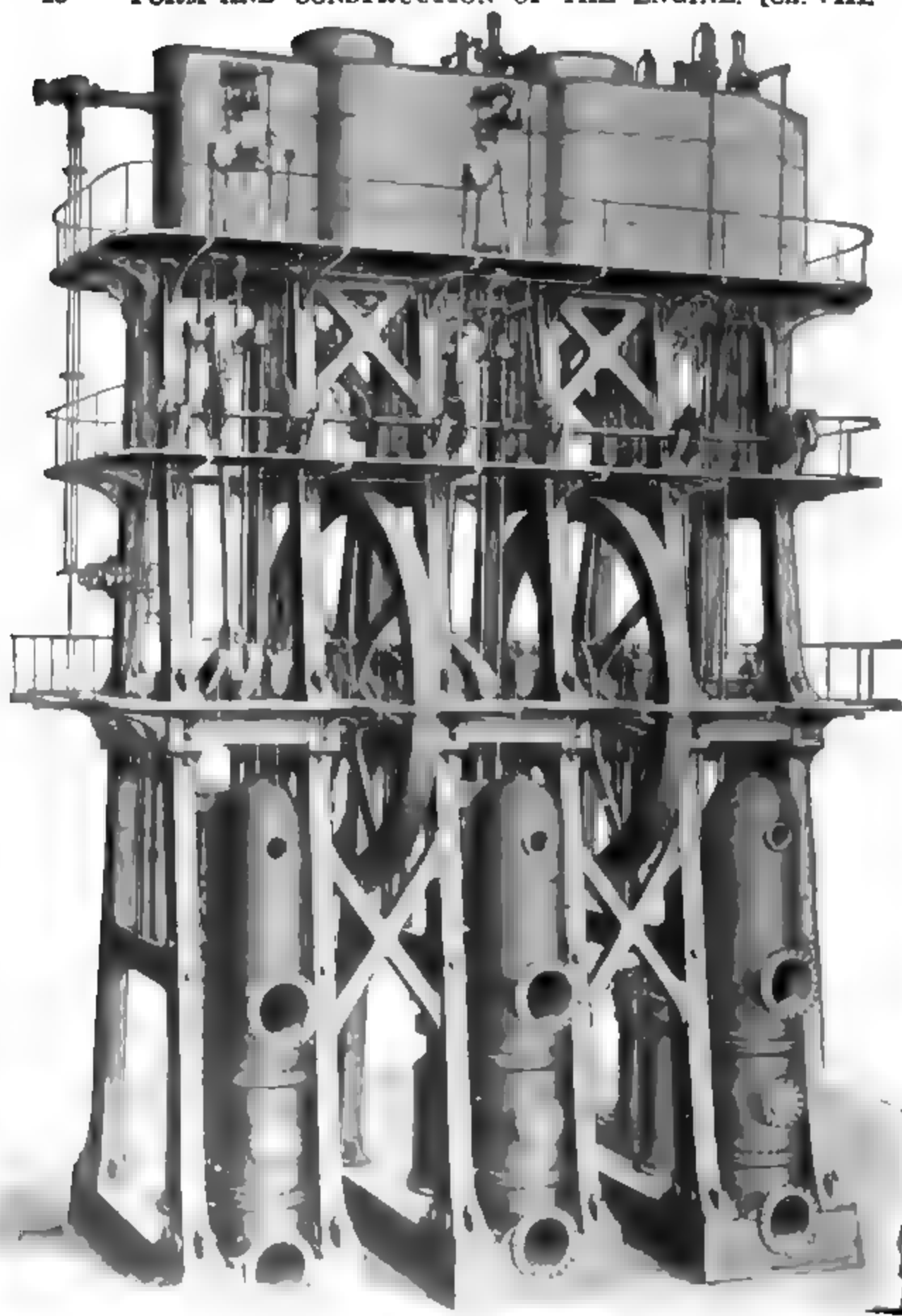


FIG. 230. - Reynolds Vertical Triple expansion Pumping-engine.





FIG. 231.—High-speed Locomotive, "Atlantic" Type. Baldwin Locomotive Works.

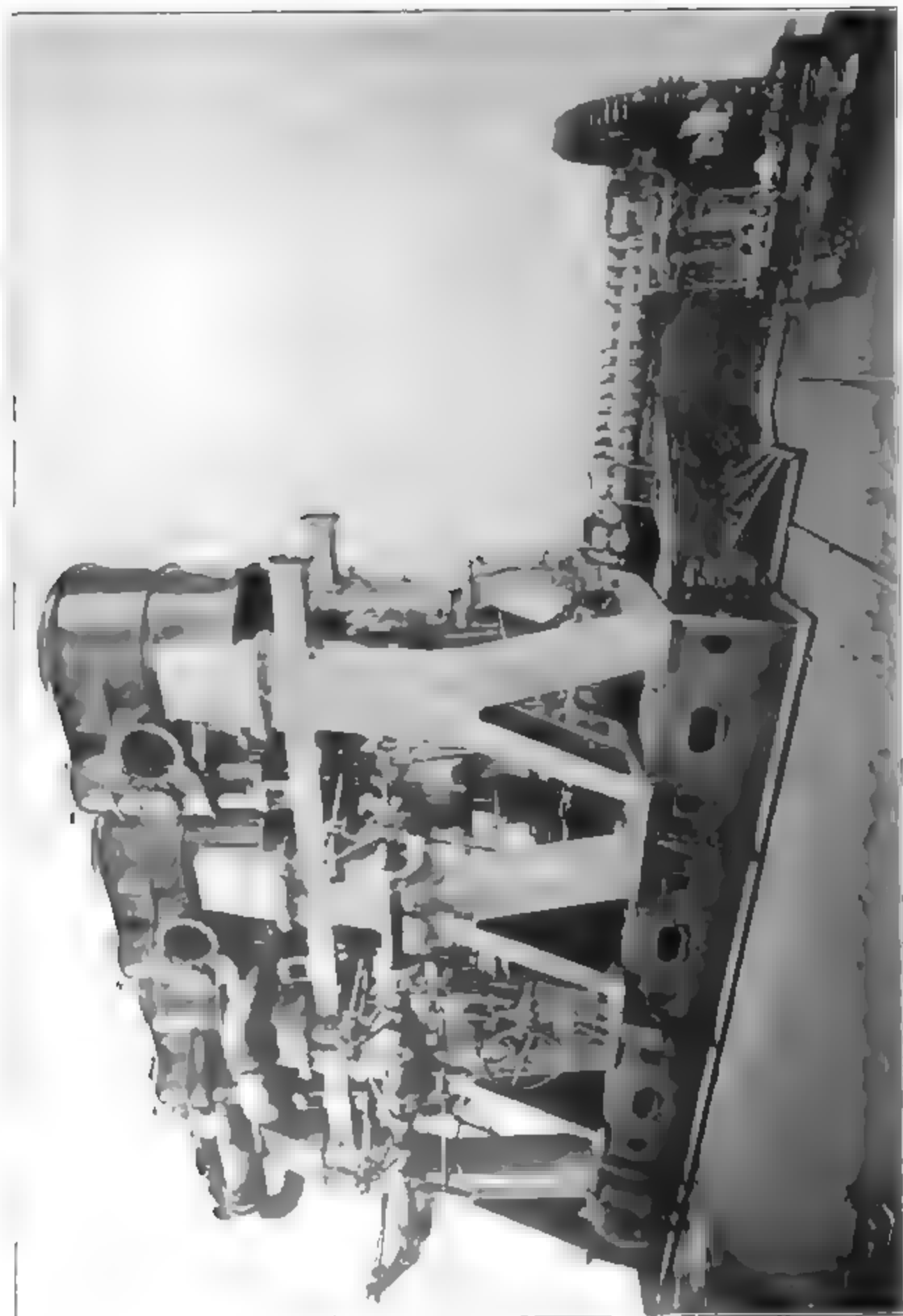


FIG. 232 Engine of Twin screw American Liner "New York" four crank triple expansion; cylinders 44", 66", 88".  
 11,000 H.P., rated 10,000 H.P., at 95 R.P.M. Built by the Cramp Company in 1902.

(q) **THE LOCOMOTIVE.**—This type of engine (or rather, of complete steam-plant) keeps itself so continually in public view that a description of its general form would be superfluous. The chief variations in the major details are found in the wheel-arrangement, the shape of the boiler-furnace, and the cylinders—the last being simple or compound. The locomotive illustrated in Fig. 231 is of the “Atlantic” type as to the wheels, has the wide fire-box or furnace, and is a simple engine. The picture is given especially to show the form of the engine proper, and will be convenient for reference when some of the less evident details are taken up.

(r) **MARINE ENGINES.**—The examples given in Figs. 232 to 235 are all of the usual type for driving screw-propellers—what is called the vertical, inverted, direct-acting engine. In this descriptive title a good deal of history is involved: the engine is “direct-acting” when it turns the propeller-shaft directly, instead of acting through a walking-beam or other intermediate gearing; and when an engine was directly applied to turning paddle-wheels, the cylinder was below, and the shaft above, hence the adjective “inverted.” From any other point of view, the arrangement with the shaft-bearings down on the foundations and the cylinders up in the air seems natural and normal.

The general form of all modern screw-propeller engines is practically the same. The variations come, first, in the number and arrangement of cylinders and cranks; next, in the form of the framework; last, in the disposal of the accessories, the condenser and its pumps, which are quite often incorporated with the main engine, but are more frequently independent, at least in the higher class of engines.

As to the cylinder-arrangement, a general idea can be got from these photographs, and one sectional drawing will be found in Fig. 259. This side of the subject is to be more fully taken up later, but a brief outline properly finds place at this point. There are some single-expansion engines yet running in ferry-boats and river-steamers, and a good many compounds in the older ships; but the engines now built are either triple- or quadruple-expansion, with from three to six cylinders and usually three or four cranks. The three-cylinder triple with three cranks

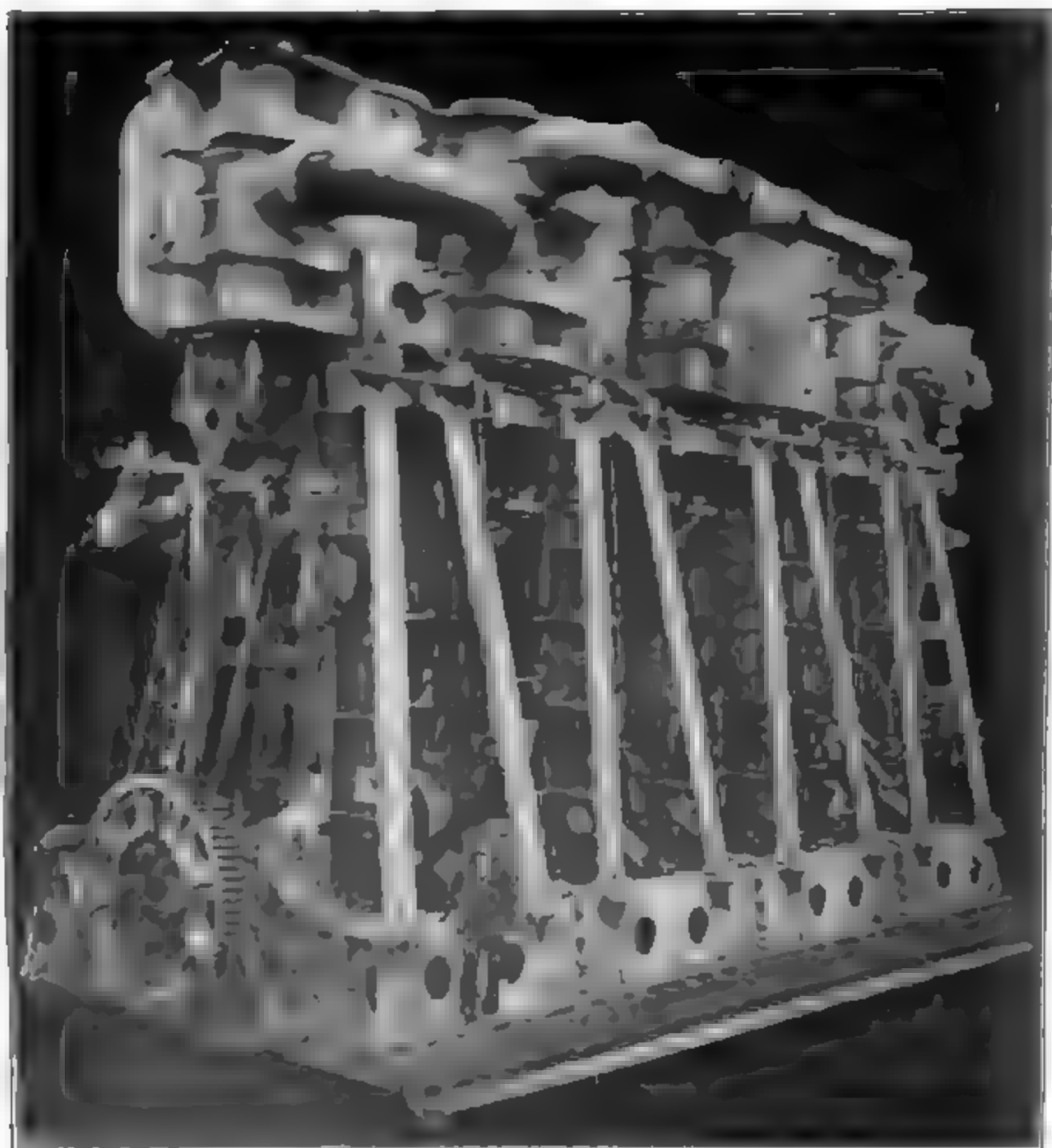


FIG. 233. Engine of Twin-screw Pacific Mail Steamship "Siberia": four-crank quadruple expansion; cylinders 35", 50", 70", 100"  $\times$  66"; each engine rated 9000 H.P. Built at Newport News in 1901.

(Figs. 231 and 239), the four-cylinder triple (two equal low-pressure cylinders Figs. 232 and 235), and the four-crank quadruple (Fig. 233) are the typical arrangements. Five-crank engines have been built; but if the quadruple has to be carried to more

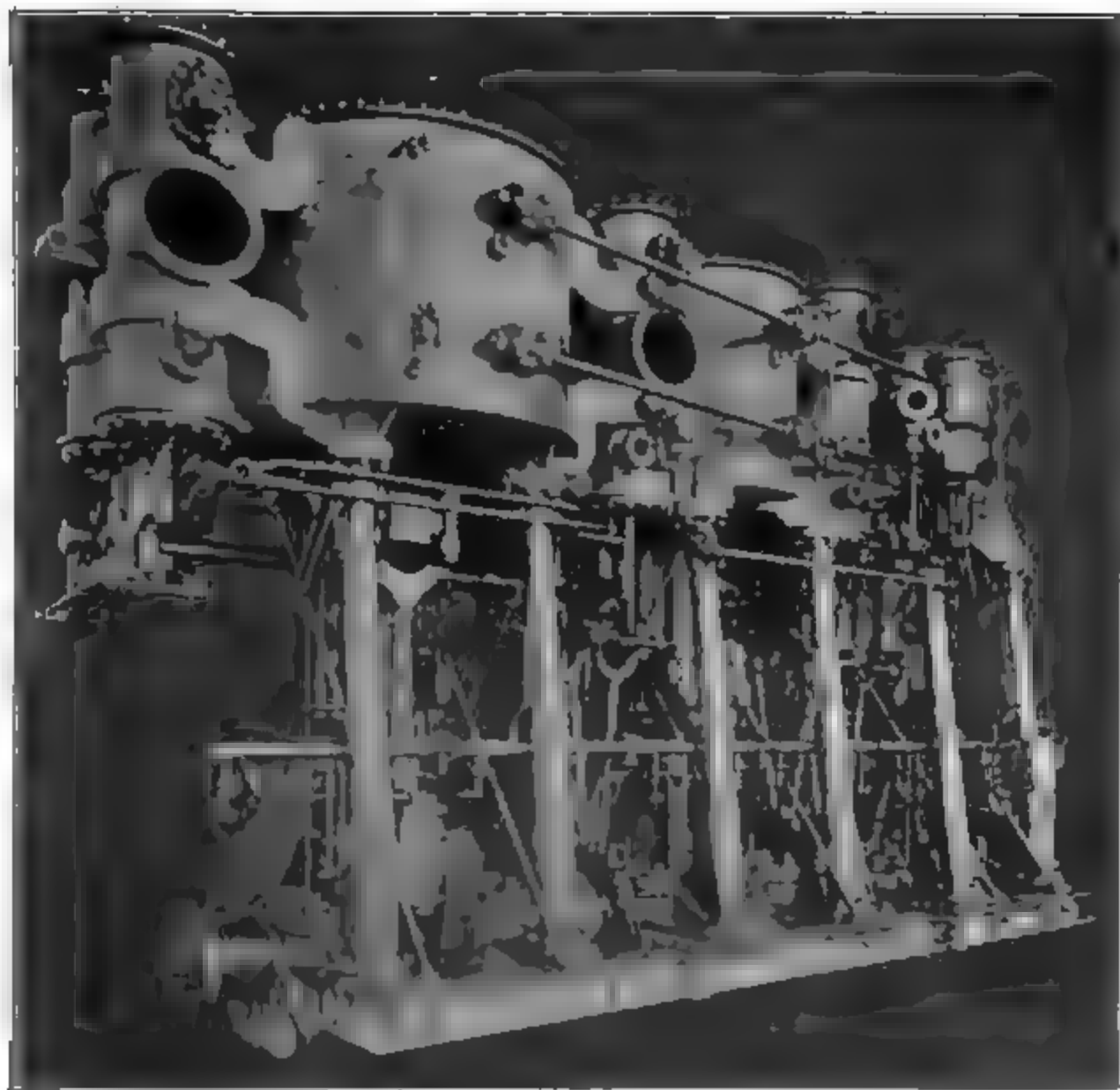


FIG. 234.—Engine of Twin-screw U. S. Battleship "Indiana"; built at Cramp's in 1895; three-crank triple-expansion,  $34\frac{1}{2}$ ", 48", 75"  $\times$  42"; mean speed of engines on official trial, 181 R.P.M.; I.H.P. of two engines, 9500.

than four cylinders (to keep the L.P. cylinder from becoming unwieldy), a tandem arrangement is usually adopted. The most prominent example of this, used on several large steamers, has two H.P. and two L.P. cylinders, an H.P. on top of each L.P., with the first and second intermediates each driving a crank.

In the matter of framework, the four common types of construction are illustrated in the examples given. Fig. 232 has double Y frames, front and back under each cylinder; in Fig. 233

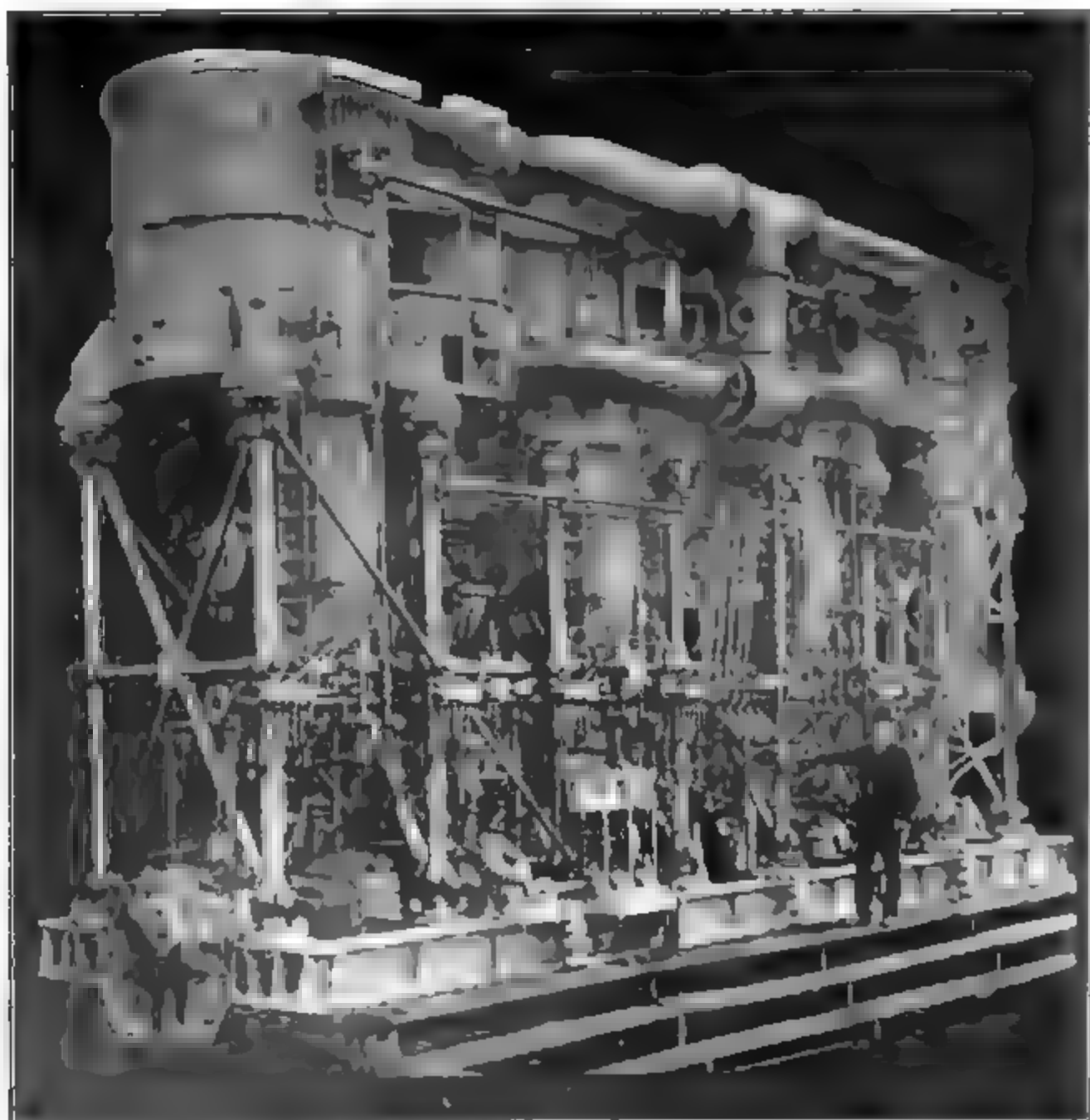


FIG 235. Engine of Twin-screw U. S. Battleship "Missouri"; built at Newport News in 1902; four-crank triple-expansion,  $84\frac{1}{2}$ ", 53", 68"-68"  $\times$  48"; each engine 7800 I.H.P.

there are four cast-steel columns in each section of the frame; in Fig. 234 the section contains a cast-iron Y housing at the back and two forged-steel columns in front; while Fig. 235 shows the all-forged frame, the war-ship type. The last has long been the standard torpedo-boat type; but is now used, in the United States Navy, for the largest engines built. The structural features of these various frames will be further discussed in the next section.

These photographs were all taken, of course, when the engine was in the erecting-shop, as neither the distance nor the illumination for a general view is possible on board ship: and thus to get the cylinders without their non-conducting covering is a decided advantage. In all four cases, the after end of the engine is in the foreground, showing the connection for the propeller-shaft: and in Fig. 232 the first section of this shaft, which carries the thrust-journal, is shown: this may be considered, and in Fig. 232 actually is, a part of the engine. A secondary apparatus, best shown in Figs. 232 and 235, is the little turning-engine, which acts upon the main shaft through two worm-gears. This is used for turning over the engine when making repairs and when warming it up before starting: the intermediate gear-shaft is always so supported that it can be swung out of "mesh" when the engine is running.

The valve-gears of these engines, all of the same type, will be referred to later: a notable feature—in which they do not fairly represent general practice—is the use of piston-valves on all the cylinders; and in Figs. 233 and 234 we have examples of four valves to one large cylinder, each taking its share in controlling the whole steam-distribution.

Besides the main steam-piping, most fully shown in Figs. 233 and 235, and which includes a number of by-pass connections from the steam inlet to the different cylinders, there are several elaborate systems of small pipes about a marine engine. First comes the drain-piping, to carry off water from all the cylinders and valve-chests, especially when the engine is standing idle; this is not visible in any of the engines illustrated; but the hand-levers for controlling the drain-cocks are shown, all grouped together near the throttle-valve wheel in Fig. 235, distributed along the front of the engine in Fig. 233. Next comes the oil-piping, simple as to its units, but with a great many of them, as best appears on Fig. 235. Finally, there is an extensive system of water-piping, whereby cooling-water can be supplied to all the bearings and to the cross-head guides.

(s) VARIOUS MODIFICATIONS OF THE ENGINE-MECHANISM.—In Fig. 236 are given skeleton-outlines of various engine-mechanisms,

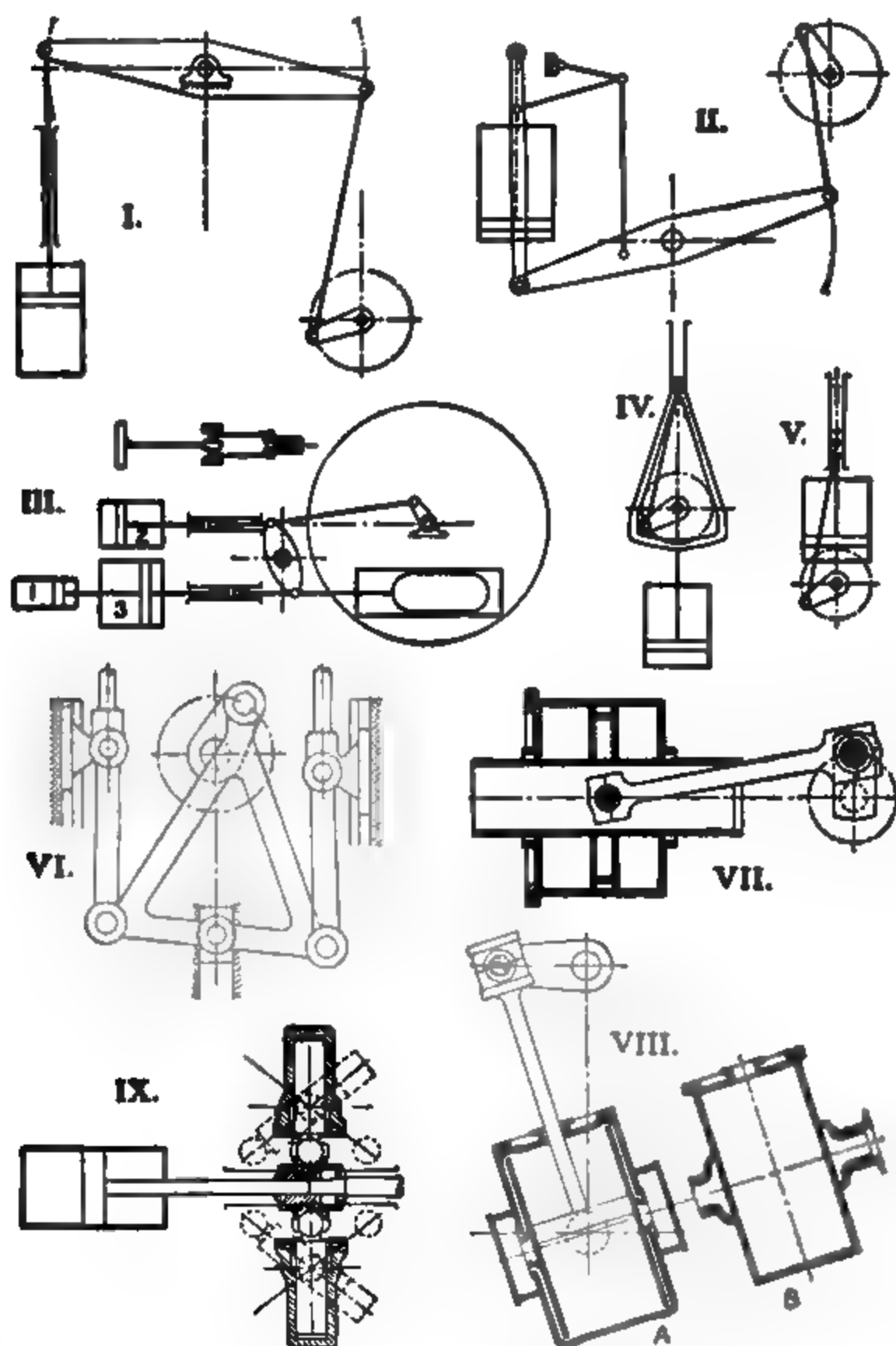


FIG. 286.- Various Forms of the Engine Mechanism.

I to III. Engines with walking-beam; IV. V. The return connecting-rod; VI. The triangular connecting rod; VII. The trunk engine, double-acting; VIII. The oscillating engine; IX. The Worthington high-duty attachment for steam-pumps.



some of them now in use, others chiefly or wholly of historic interest.

The earliest engines built, for mine-pumps with a heavy pump-rod running down the shaft, were of the walking-beam type; and in the application to driving machinery, to common pumps, to blast-furnace service, and to the turning of paddle-wheels on steamers, this type, as outlined at I., was very largely used. A modified form for marine service is the side-lever engine at II. In I. the upper end of the piston-rod is guided by a cross-head, as is now universal practice; in II. a "parallel-motion" of jointed links is used: this, in various forms, was standard practice in the early days of steam-engineering.

In connection with this matter of the walking-beam, the mechanism of the Holly pumping-engine, Fig. 227, is outlined at III.; the plan view shows how double connecting-rods are used from the cross-head to the beam, and how the latter is made double, so as to give room for the main connecting-rod at the top and to clear the pump-rod at the bottom: incidentally, the cylinder arrangement in a triple-expansion engine of this type is shown. Some designers of pumping-engines have made a very free use of the oscillating lever in the main mechanism, frequently giving it a triangular or "bell-crank" form, and securing the advantage of a longer stroke and higher piston-speed for the engine than is allowable for the pump.

The return connecting-rod arrangement, in two forms, is sketched at IV. and V. The "steeple" engine, IV., belongs to early marine practice; it is intended rather to get a large cylinder into a comparatively low space beneath the paddle-shaft than to shorten up the machine as a whole; and the heavy triangular frame which carries the piston-rod around and past the connecting-rod is a bad feature. The simple return-rod engine, used for the steam end of air-compressors, and also in marine practice when space is crowded, is skeletonized at V.

The triangular connecting-rod, VI., has been used by several designers (or inventors): for the description of an unusually complex engine of this type, see *Engineering*, 1899, II., page 580.

Two arrangements by which the engine is greatly shortened

are next shown; the trunk-engine, VII., combines piston cross-head in one, as already explained in connection with 207. The oscillating engine, shown at VIII., is a kinetic variation from the common type, its frame corresponding to the connecting-rod in the usual mechanism. Steam is conducted to and from the cylinder through the trunnions upon which it turns, so that the engine properly belongs to the era of low-pressure and moderate speed. To transmit the valve-movement from the crank-shaft to the valves (on the oscillating cylinder) is an interesting problem.

Sketch IX. shows a special mechanism used on the pump engine in Fig. 228—the Worthington high-duty attachment. Hydraulic plungers from two oscillating cylinders bear upon the cross-head; in the first part of each stroke these plungers are forced into their cylinders, resisting the movement of the piston-rod; in the latter half, they assist the driving-force. The cylinders are in constant communication either with the discharge-pipe from the pump, or with a special small accumulator, indirectly connected to the force-pipe. These compensating cylinders induce a force-action very similar to that of the inertia of the rotating parts in a crank-controlled engine, and perform the function of a fly-wheel, by storing up the excess-work of the steam-pressure at the start and returning it later, thus making it possible to use quite a fair ratio of expansion in each cylinder.

(I) THE "LAYOUT" OF AN ENGINE.—The general description of an engine, besides covering matters of service, position, arrangement of cylinders and cranks, type of valve-gear, etc., should set forth also certain particulars as to the arrangement of the mechanism and the direction of running, according to the following terms:

*Right and Left.*—It has been the custom to call a horizontal engine "right-hand" if, when we stand back of the cylinder face toward the shaft, the wheel is at the right. A decidedly logical scheme is got by reversing this practice, calling the opposite the wheel and valve-gear the front of the engine, going by the right or left position of this front. Then the engines in Figs. 209, 210, 212, 213 are right-hand, and those in Figs.

206, 211, left-hand; and in a cross-compound engine, as Fig. 216, this makes the right-side element right-hand, the left-side, left-hand. In a center-crank engine the distinction is less marked and of less importance; it is best simply to specify on which side the governor is placed, and on which side the generator, in a direct-connected unit.

*Over and Under.*—A horizontal engine runs over if it makes its forward stroke—the piston moving toward the shaft—while the crank traverses the upper part of its circle: in which direction of running the pressure of the cross-head upon the guides is normally downward. The characteristic position of the engine mechanism, in Fig. 110, belongs then to a right-hand engine, running over, and with right-hand or clockwise rotation when viewed from the front.

In a vertical engine, the front side is properly that toward which the crank-pin travels when traversing the upper part of its path. With a symmetrical frame, as in Figs. 217, 218, 232, and 233, there is little in the structure to mark “front” and “back”; but with the type of framework in Figs. 202 and 234, and with the one-sided slipper type of cross-head, the open side is naturally the front, and is the one toward which the crank turns over when the bottom of the cross-head is pressing upon the guide.

If a vertical engine is symmetrically loaded, as with a generator at each end of the shaft or with one in the middle, no clear distinction as to right and left rotation can be drawn. With the marine engine, if we stand back of the engine and face forward, right-hand rotation corresponds to forward running with a right-hand screw; and another reason for calling this a right-hand engine is that its front will be at the right, from the view-point specified. Then with twin-screws turning outboard (outward at the top), the starboard (right-side) engine will be right-hand, the port engine left-hand. Whether it is better to have the screws turn outboard or inboard is a point upon which marine engineers differ with no overwhelming arguments for either arrangement.

(u) SPEED OF ENGINES.—This is measured in two ways, by the rotative speed or the revolution per minute, and by the piston-speed or the distance in feet travelled by the piston in one minute.

The data in Tables 41 A and 41 B will give a good idea of the usual range in stationary practice.

TABLE 41 A.    DATA FOR HIGH-SPEED ENGINES.

Stroke.	R.P.M.	F.P.M.
12".....	260 – 300.....	520 – 600
16".....	210 – 250.....	560 – 667
20".....	180 – 210.....	600 – 680
24".....	150 – 180.....	600 – 720

TABLE 41 B.    DATA FOR ENGINES OF THE CORLISS TYPE.

Stroke.	R.P.M.	F.P.M.
24".....	85 – 125.....	340 – 500
30".....	80 – 115.....	400 – 575
36".....	80 – 110.....	480 – 660
42".....	75 – 100.....	490 – 700
48".....	70 – 90 .....	560 – 720
60".....	60 – 75 .....	600 – 750

It is at once apparent that the distinction between the two classes of engines is found chiefly in the R.P.M. It will be noted further that the range in piston-speed with any particular stroke is greater in the second table than in the first; and for the Corliss engine, much more than for the smaller type, the lower set of limits represents what has long been usual practice, the upper set stands for more recent installations.

Accepting for a convenient basis of comparison this usual range from 500 to 750 ft. per min., as set forth in the tables, a wider view of practice in this matter will yield the following results:

The lowest speeds are found in steam-pumps like Figs. 225 and 226, which are built with the stroke-length varying from 4" to 24": for 6" and less, the proper number of "revolutions" is usually set at 75, or of single strokes at 150, giving 75 F.P.M. or less; for 8" and more, the limit of piston-speed is fixed at 100 F.P.M. Large, long-stroke pumps without fly-wheel control, like Fig. 228, have higher piston-speeds, rising to 160 F.P.M. Pumping-

engines with fly-wheels come next, the piston speed usually lying between 120 and 250, sometimes rising to 350, and in extreme cases, with a very long stroke, exceeding 400 F.P.M. The smooth running of any pump is very much a question of the uniform flow of the current of water: it is possible to have pulsations set up which will bring very severe strains upon the body of the pump and all its working parts. Other things being equal, it would appear that three pump-cylinders connected by cranks at  $120^\circ$ , as in Figs. 229 and 230, ought to give the most uniform discharge possible in any arrangement of the sort, having a slight advantage over the quarter-crank duplex pump.

As to blowing-engines, we will take the example in Fig. 224, where the range of control is from 250 to 500 F.P.M., to be typical. For air-compressors with self-closing valves, as in Figs. 222 and 223, having a stroke from 12" to 30", the piston-speed ranges from 320 to 400 or 450. In larger machines with mechanically-moved valves, the speeds can be much greater, running up to 600 or 700 F.P.M.

In large power-house engines with positive valve-gear, as typified by Fig. 218, the piston-speed rises to 900 F.P.M.: and this is about the limit for stationary engines. Where space and weight are not the most exacting conditions, it will generally pay better to get a machine which will not have to be worked quite up to the limit.

In transportation service, both marine and locomotive, the conditions just named are, however, of primary importance. Slow freight steamers with low power have piston-speeds from 500 to 700: but in large passenger steamers and in warships at full speed the pistons travel 900 to 1000 F.P.M.: typical cases being, 60" stroke by 90 to 100 R.P.M., 72" by 80 R.P.M., for large liners; and 39"  $\times$  160, 42"  $\times$  145, 48"  $\times$  120, for cruisers and battleships. In torpedo-boat destroyers the piston-speed runs up to 1200 F.P.M., 18" stroke by 400 R.P.M. being a very common size. The greatest speeds are formed in the locomotive: in usual working the piston travels from 500 to 1200 F.P.M., according to the class of service; but in the very fastest running, with high-speed locomotives of usual proportions at from 70 to 75 miles per hour, the piston-speed

rises above 1400 F.P.M. Thus the locomotive in Fig. 231 has made 45 miles in 37 minutes, or 73 miles per hour, hauling its usual train; with 76" drivers, this gives 323 R.P.M; and with 26" stroke, the piston-speed is just 1400 ft.

#### § 42. The Framework of the Engine.

(a) THE COMPACT, HIGH-SPEED TYPE.—The bed of one center-crank engine is quite fully shown in Figs. 2 to 5; another is given in Fig. 237; and a third, of the side-crank type, in Fig. 238, which is supplemented by Fig. 263: with these three examples, the class is very sufficiently represented. To a large degree, the drawings are expected to speak for themselves; but it may be well to call attention to some points of resemblance and of difference in the designs, so as to suggest the requirements that must be met, and illustrate the manner in which they are satisfied.

In each case, the engine-bed is to rest upon a cast-iron sub-base; its bottom outline is therefore given a simple rectangular shape, except where it swells out under the bearings; this latter effect being relatively greater in the side-crank design, where the main base is narrow.

As to the manner of fastening the cylinder, we have two cases of inside bolts, one of outside bolts. In these engines, as always, the agreement of the axis of the cylinder with that of the bed is determined, not by the bolts, but by a projection on one piece which fits neatly into a recess in the other—whether the projection be a shoulder on the cylinder-head as in Fig. 237, or be at the stuffing-box as in Figs. 3 and 263. In good construction, the bolts never act as dowel-pins, do not have to be accurately fitted, and are not subject to shear.

At the guides our three examples show three typical forms, suited to different types of cross-head. Fig. 237 has four-bar guides, for a wing cross-head, shown in detail by Fig. 298. Figs. 3 and 4 show a broad flat surface for a slipper cross-head, formed right on the main casting, with side-bars to keep the cross-head from lifting. In Fig. 238 the guides are bored, to receive a cross-head of the block or box type.

In Figs. 3 and 238 the bottom of the casting forms the lower guide, then slopes toward the crank-space and toward the cylinder-connection, with, of course, very considerable differences in detail.

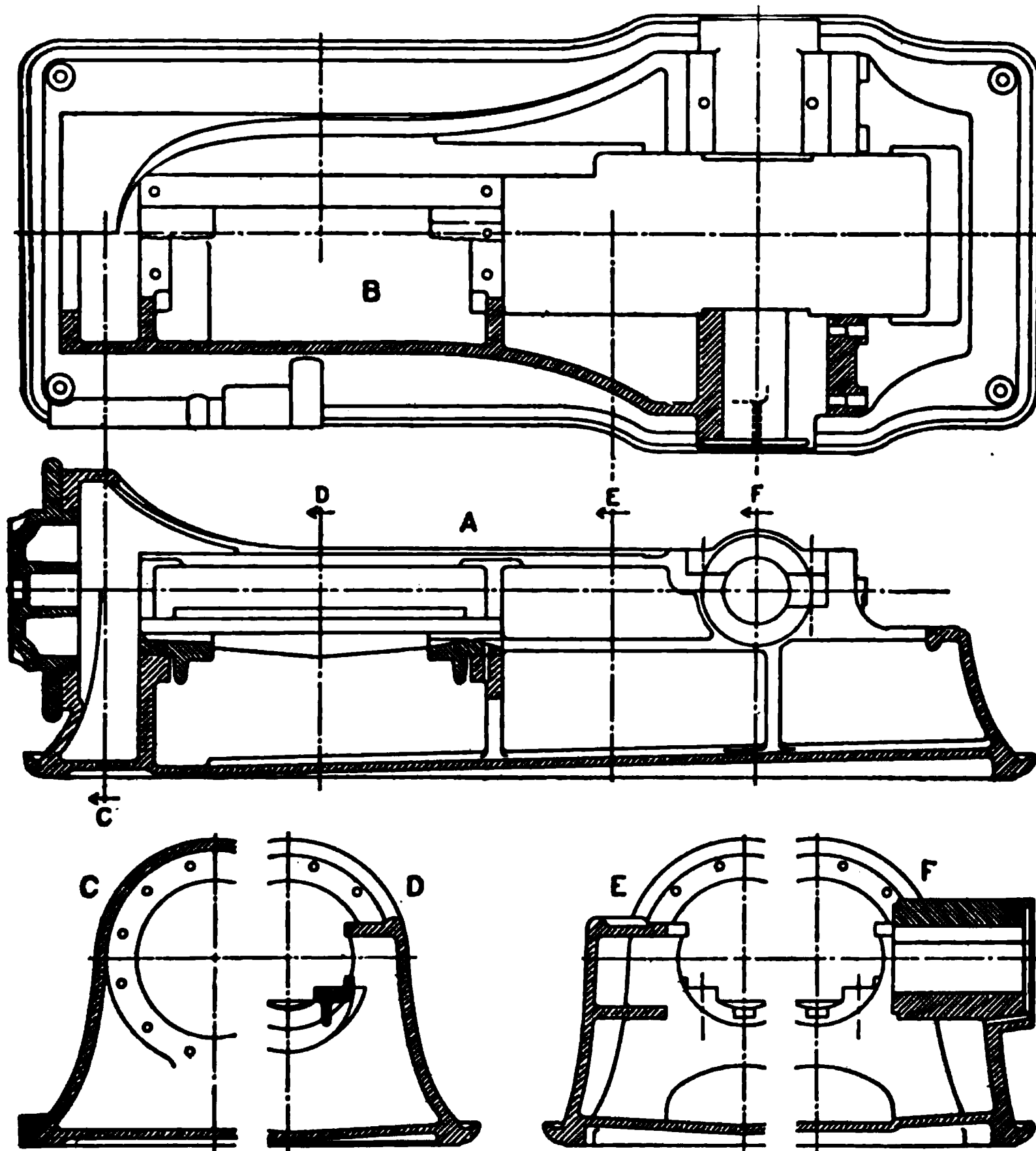


FIG. 237.—Bed of 15" by 14" American-Ball Engine, Fig. 201.  
Scale 1 to 24.

Fig. 237 shows another shape altogether, the bed being closed at the bottom by a nearly flat floor. With any stationary engine, it is highly important that oil from the machine be not allowed to soak into the foundation; in the type under discussion, whether

horizontal or vertical, the bed-casting is made so as to catch and contain all oil, even to the extent of adding outside troughs as on Fig. 237 and at the bearing on Fig. 236. With many engines

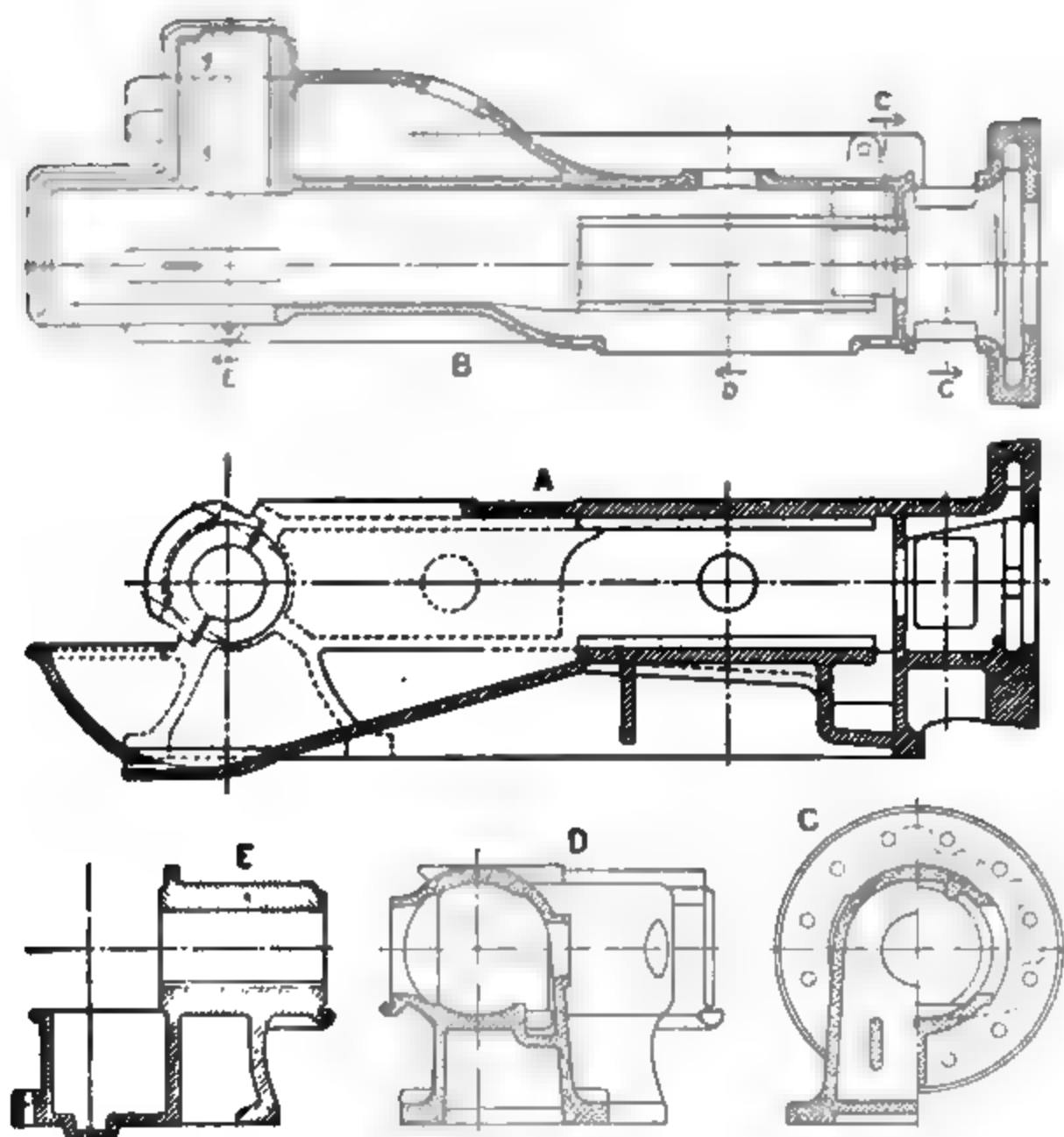


FIG. 298.--Bed of 16' by 16" Fleming Four-valve Engine, FIG. 215.  
Scale 1 to 24.

of the more open type, sheet-metal troughs and floor-guards are necessary.

The oil-catching spaces are always provided with drains from their lowest points. Note in Fig. 3 how an oil-reservoir is formed in the sub-base at the bearing end. Aside from the need of a par-



tition between the main enclosed space and the chamber in front of the stuffing-box when splash lubrication is used as in Fig. 238, there is good reason for the same separation with jet-lubrication as in Fig. 237; because the water dripping from the stuffing-box ought not to be allowed to mix with the oil from the bearings, especially when this oil is kept in rapid circulation by an oil-pump. The form of the guards or shields which enclose the working parts can be seen upon, or inferred from, the general illustrations.

The remaining part of importance is the bearings. These are to be considered in detail when the shaft is taken up. For the present it is enough to remark that the Ames engine has a cast-iron bushing, all in one piece, and lined with babbitt: the other two have plainer, solid bearings, with babbitt lining of course; but we note the typical difference between the arrangement with a top cap and adjustable quarter-box in Fig. 237, and that where the bearing is divided almost at right angles to the principal line of stress, in Fig. 238. In the latter case, adjustment can be made, if needed, by using liners or shims beneath the cap at the joint, and removing them when wear must be taken up.

Bosses or seatings for the support of the valve-gear are placed on the engine-bed as needed, without affecting the general lines of its design.

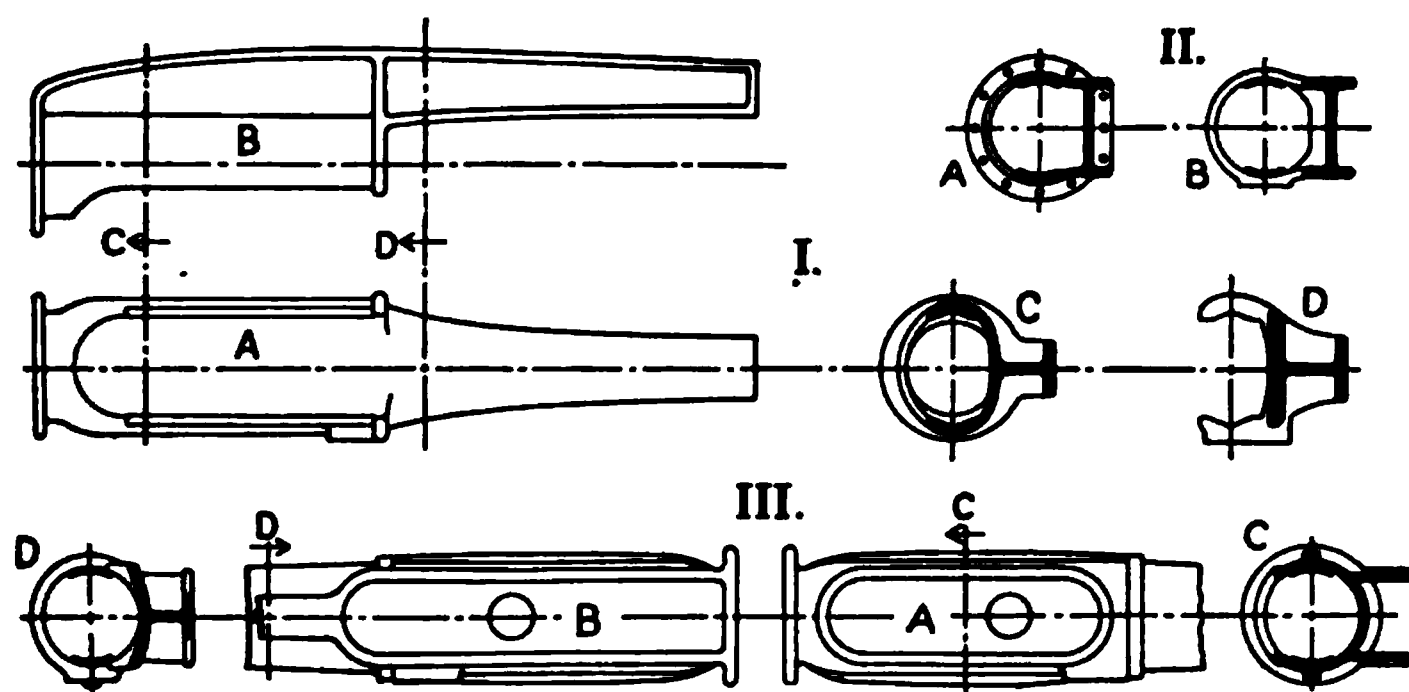


FIG. 239.—Various Girder-frames for Corliss Engines. Scale about 1 to 96.

(b) CORLISS ENGINE-BEDS.—The older type of open girder-bed is illustrated in several forms in Fig. 239: I. has the common T-

bar section, as shown especially at D; that is, it has the flange stiffening rib, as in Fig. 210; II. is a less usual form, with I-beam cross-section; while in III. the stiffening rib is the double back of the guides, or something like a box-girder is used. The relation between the several views of I. is indicated by the cross-lines and arrows; in II. only the cross-sections A and B are given, and these correspond with C and D of I.; in III. is the front view. We have here one example of V guides, another in Fig. 240; with bored guides on one of these girder frames, it is quite usual partly to enclose the guides as in Fig. 241 by running a half-ring from one to the other at the front

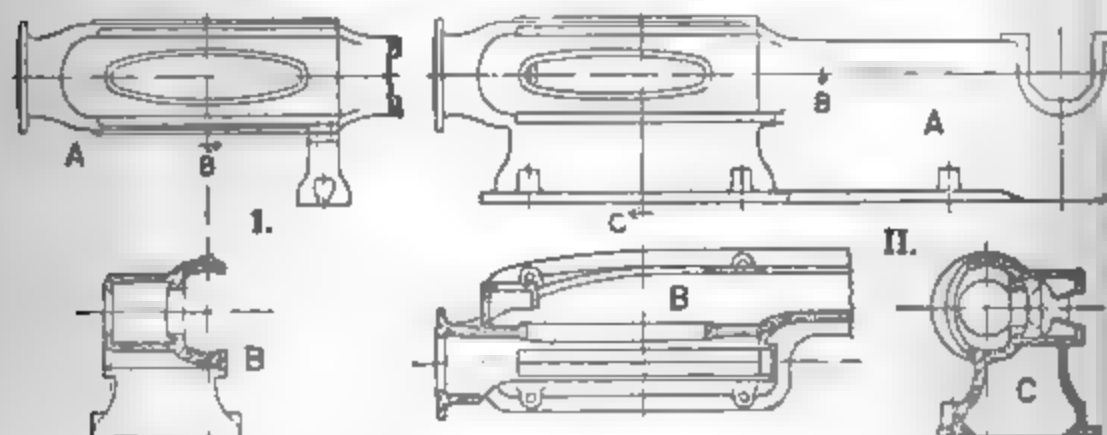


FIG 240. An Instance of Evolution in the Bates-Corliss Engine.

An interesting example of development from the older girder bed to the now prevalent full-base type is given in Fig. 240: I shows the cross-section of the box-girder, which is essentially the same in both designs, and is further brought out by II. B: I perhaps even better than A, shows how the full base is developed by a simple and harmonious addition to the original design.

Fig. 241 is given with the intention of bringing out very clearly the form of the casting, showing all the hollow parts and the form and location of the internal stiffening-ribs. The most intricate coring work is at the collar around the front end of the guide where the cross-section C is taken. It is of interest to note that the main web marked W is continuous, and almost straight, from the main-bearing to the cylinder-seating. The hole back of the guides, at section B—as likewise in Fig. 239 III.—is for the purpose of getting at the inner end of the wrist-pin, when it is to

removed from the cross-head. A detail of the bearings of this engine will be found in § 45, at Fig. 320.

(c) VERTICAL STATIONARY ENGINES.—Only one frame of this type is illustrated in detail, because so much can be seen by, or inferred from, the external view, as in Figs. 218, 219, 224, etc. The fact that the function of carrying the engine is taken by the bed-plate—the supporting forces acting in the same line as the working forces—makes the casting simpler than in a horizontal engine. The example given in Fig. 242 is rather more complex

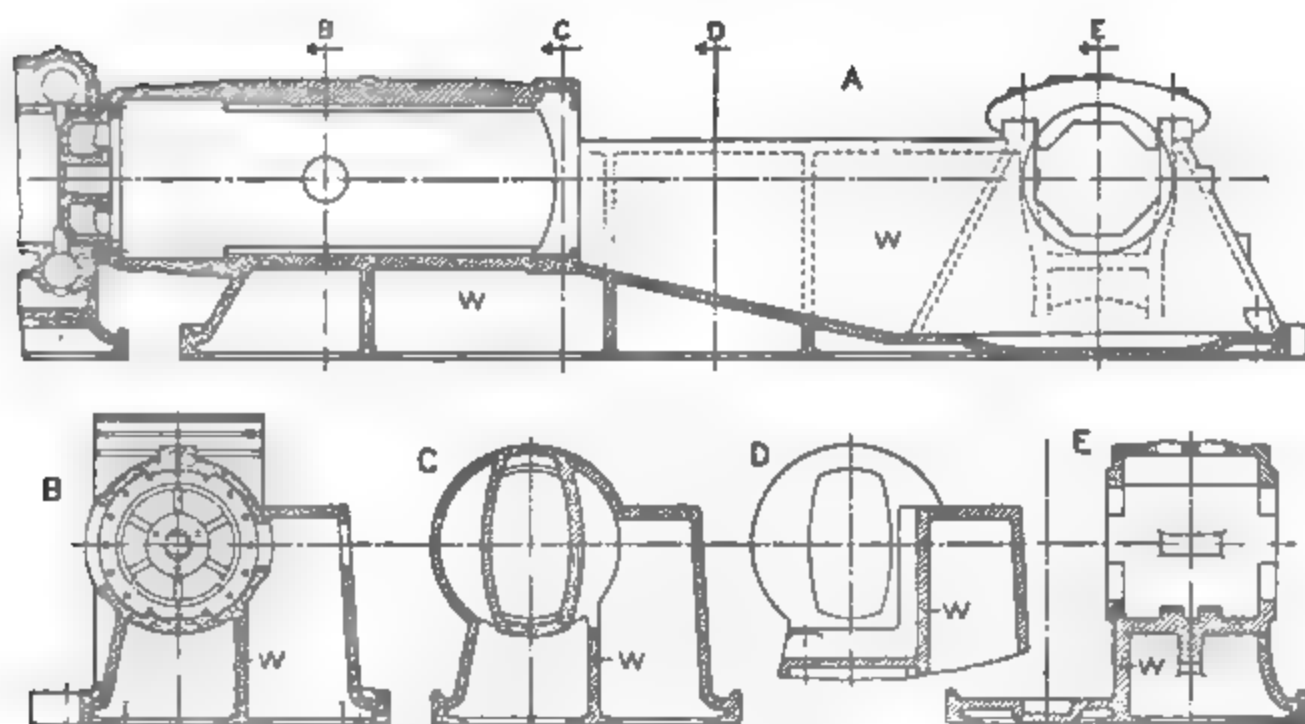


FIG. 241.—Frame of 26" by 48" Murray-Corliss Engine, Fig. 218.  
Scale 1 to 60.

than usual, the frame consisting of three pieces, above the bed-plate. The following description will supplement the drawing:

This engine-frame, of the very usual A-shaped form in its general outline, is made up of two principal parts, the housing 1 and the guide-barrel 2: further, the housing is here made in two pieces, bolted together above the crank-arch. The advantage of this construction is, that one of these halves can be removed, making it a very easy matter to take out the shaft; the other half being sufficient to carry the mere weight of the engine, which is small in comparison with the steam-forces. View CDEF bears

an important part in showing the detail of the housing, while (H) serves a like purpose for the guide-barrel. The latter

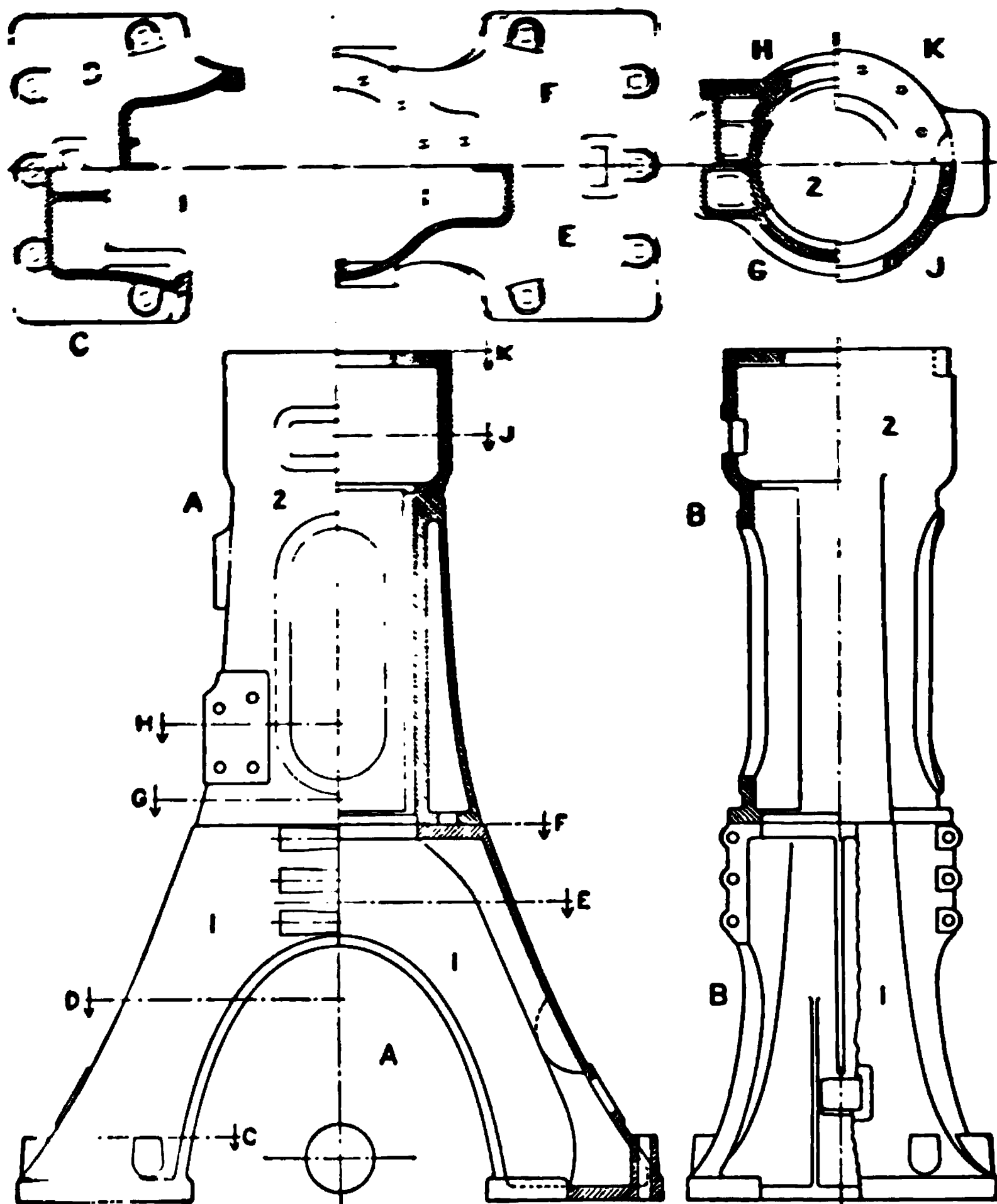


FIG. 242. Frame of Vertical Cross-compound Buckeye Engine, 22' and 42' by 27"; see Fig. 217. Scale 1 to 36.

piece is somewhat complicated by the cored-out hollows in the bracing ribs back of the guides: otherwise it is a simple, straight-

forward casting. The only detail omitted in this drawing is certain lugs which are formed upon the casting to support the platforms or galleries seen on Fig. 217.

The bed-plate of this or any other vertical stationary engine is a comparatively simple piece. It is, or should be, always made with a bottom-web running under the crank-pit, so as to keep oil and water off the foundation. The photographic views already given show clearly wherein it differs from the marine-engine bed-plate, which is now to be further illustrated by drawings.

(d) FRAMES OF MARINE ENGINES.—The first example, Fig. 243, is from a smaller engine of the same general type as Fig. 235; and the cylinders of this engine are further illustrated in Figs. 259 and 260. View A takes in only one division of the frame, that which supports the high-pressure cylinder, and gives also a half-section of the cylinder; while B includes a section through the valve-chamber. The cylinder-arrangement is given in Fig. 259, which shows how the three cylinders are flanged and bolted together. In large engines, however, the cylinders are not thus rigidly connected, because their expansion when heated would throw the stroke-lines out of parallel, and because the cylinders would be subjected to awkward stresses. Constant distances between the cylinder-axes are preserved either by longitudinal struts bolted between the frames at the top, as in Figs. 232 and 233; or by having tie-rods which take hold of stout lugs on the cylinders, as in Figs. 234 and 235. The bed-plate, Fig. 243 C, is of cast steel, and the section belonging to one cylinder, with its two shaft-bearings, is cast in one piece.

An interesting example of the frequently used composite frame—cast housings at the back, steel columns in front—is given in Fig. 244. The cranks are arranged very nearly as in Case B, Figs. 183 and 185; and further, in order to diminish the free turning or tipping moment, the two cylinders at each end are crowded as close together as possible. This throws two sets of valves together in the middle, and separates the two end groups by the full distance needed for these valve-chambers—giving the layout seen in Fig. 244 C. Each pair of cylinders is supported by two cast-iron Y columns and three steel columns; of the latter,

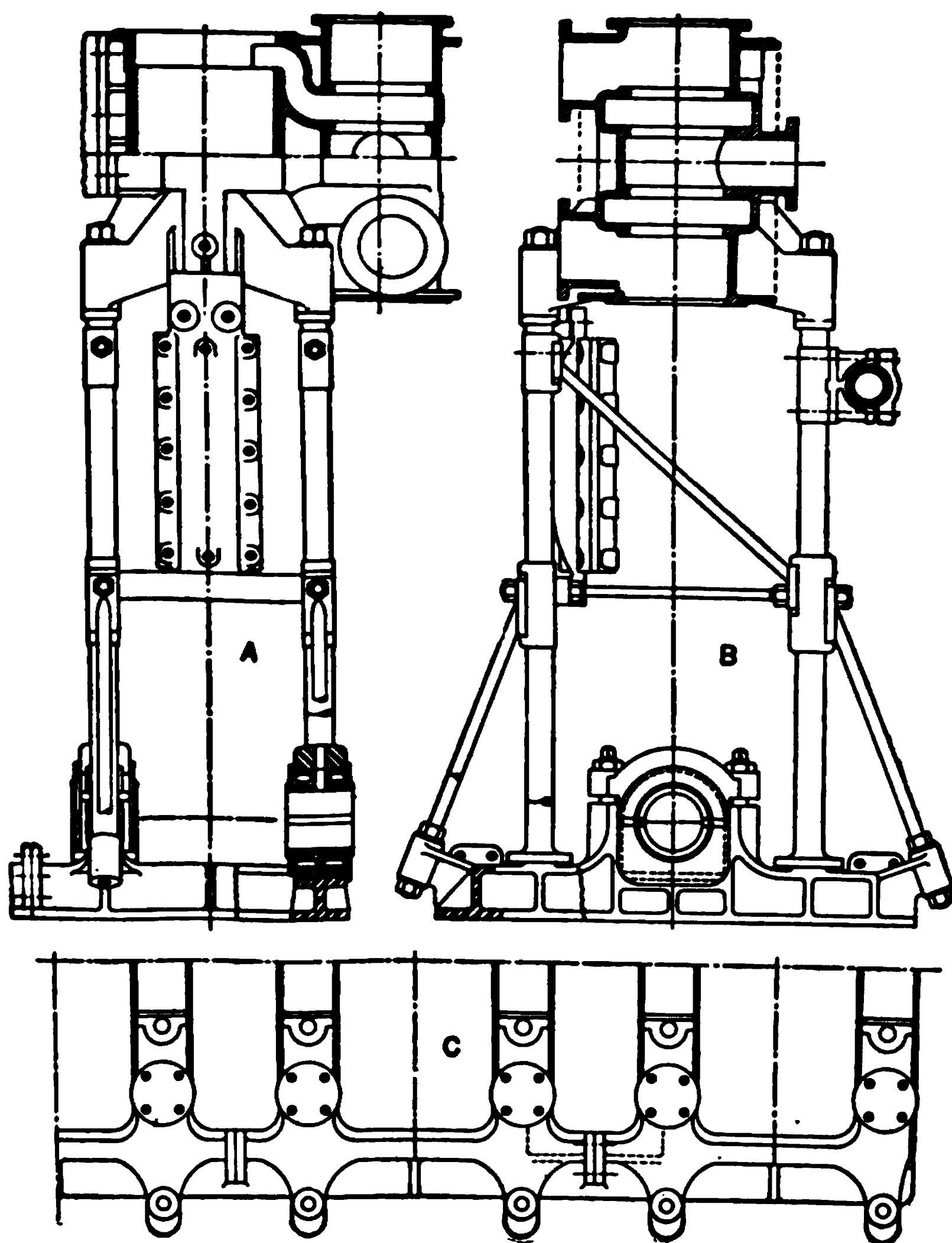


FIG. 243.—Frame of Gunboat Engine, U. S. Navy; see also Fig. 259.  
Scale 1 to 24.

the one in the middle (at the left in A) is heavier than the others, and is also made taller so as more conveniently to support the short brackets on the crowded sides of the cylinders. It is here

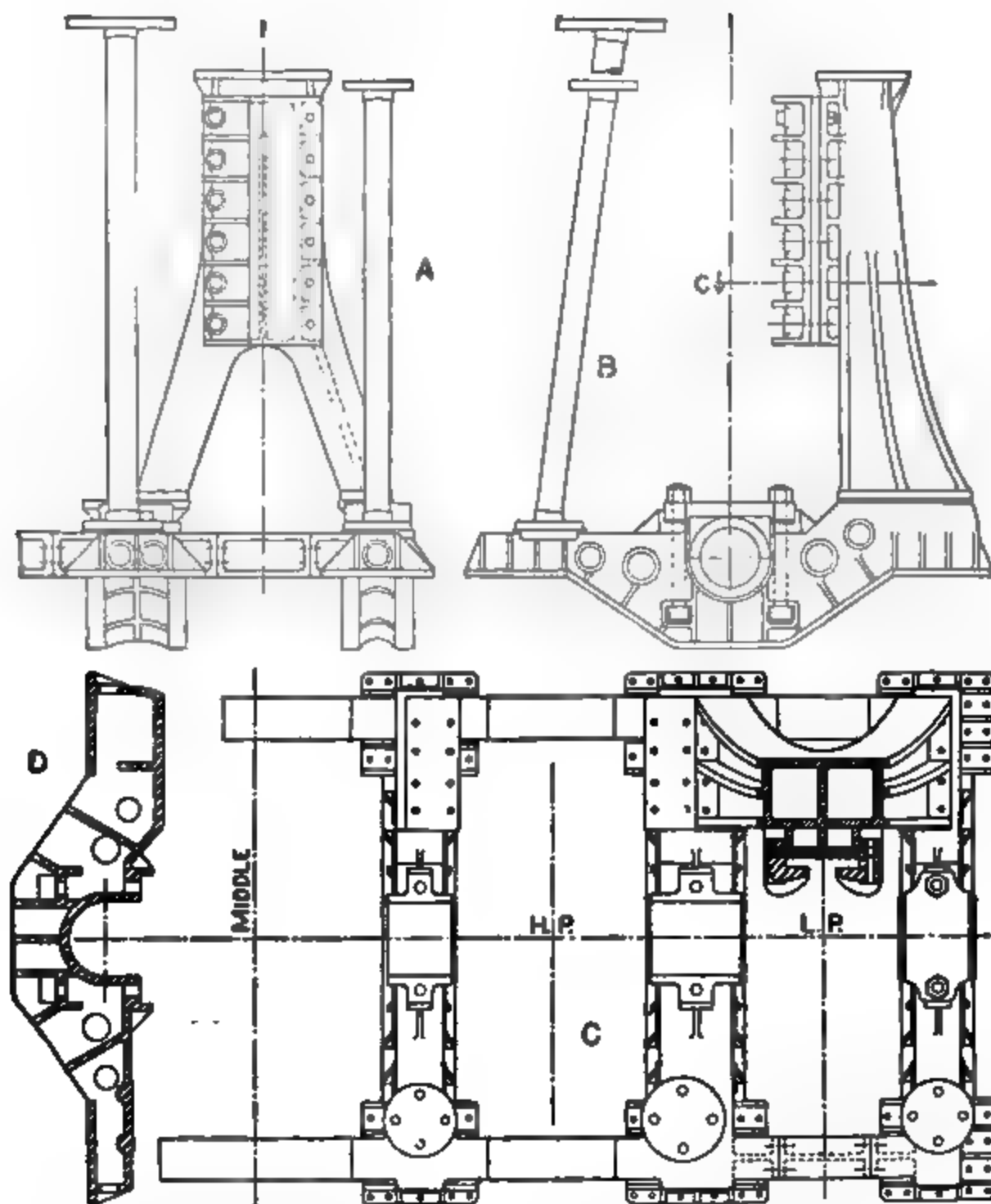


FIG. 244. — Frame of Engine for British Battleship "Triumph"; four-cylinder triple-expansion, 29", 47", 54" and 54" by 39". *Engineering*, 1904, I, page 90. Scale about 1 to 56.

evidently considered easier to machine the end-flanges oblique to the shanks of the columns than to plane the seatings exactly at



FIG. 245. Frame of Engine for Small Passenger Steamer, cylinders 17", 17", and 40" by 34".  
Scale 1 to 48.

The main-bearing frame is a cast-iron frame which makes the connections rather large—so



that this engine must depend, for rigidity, very much upon the foundation which is built into the framework of the ship. Fig. 244 is, in this respect, not representative of usual practice: much stronger frames of the same general type are seen in Figs. 234 and 235; while the deep bed-plates in Figs. 232 and 233 are even more rigid.

Fig. 245 shows the frame of a moderate-sized triple-expansion engine for a small passenger steamer, the most striking feature being the incorporation of the condenser-body into the engine-frame—an arrangement frequently used in the smaller and slower merchant steamers. This whole frame is of cast iron, except the single steel front column under each cylinder; and here the main bed-plate is all in one piece, fastened to the condenser with six flanged joints. The scale is too small for showing details of the bearings and of the condenser: on the latter are indicated, however, the exhaust-inlet at E, the air-pump connection at P, the water-connections W, W, and several hand-holes marked H. For the rest, the detail of the casting is quite clearly given by the sectional views.

(e) LOCOMOTIVE FRAMEWORK.—This type of construction is sufficiently represented by the example in Fig. 246, which is of simple form, with no special or peculiar features. The engine-cylinders are each cast in one piece with half of the saddle 2, as best seen at C and D; in A the cylinder in the foreground is cut away, and we see a section of the neck between cylinder and saddle, with the passages marked, S for live steam, E for exhaust. The “frames” proper, 3, 4, are forgings made by welding together heavy wrought-iron bars, though the use of steel castings for these parts is becoming more and more prevalent. The front frame 3 is securely bolted to the saddle, besides being held by shoulders upon the bar; and is fastened to the main frame 4 by a strong splice. Piece 4 contains the “jaws” for the axle-boxes, with panels between them, each jaw being closed at the bottom by a pedestal-brace of the bolt-and-thimble form. A “ten-wheel” locomotive has three driving-axles; the place for one of these is cut out in the figure, on account of limitations of space.

Without going into a description of the cross-framing and the

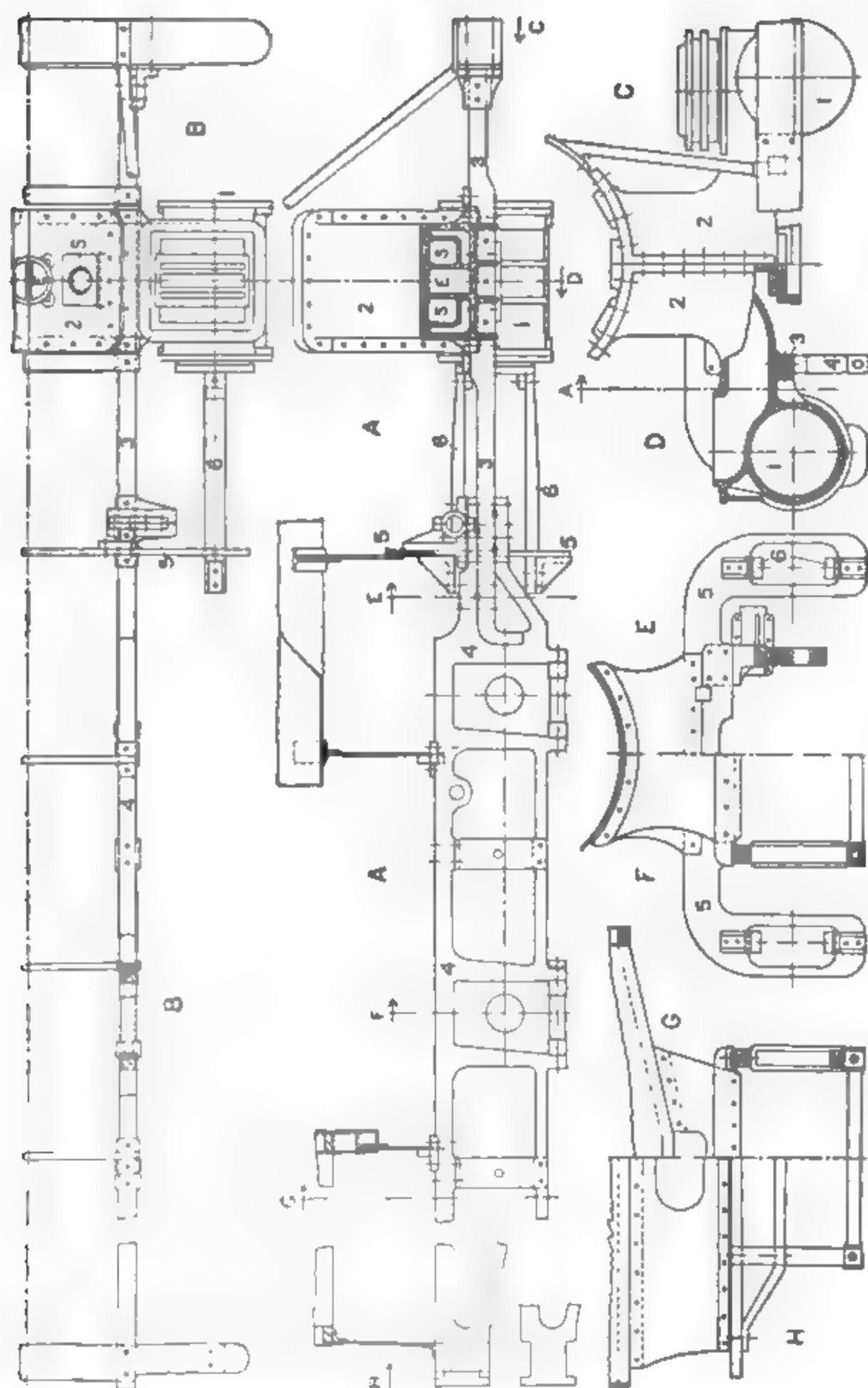


Fig. 246.—Frame and Cylinders of Simple Locomotive, "Ten-wheel" type. Scale 1 to 48.

1. Cylinder. 2. Saddle. 3. Front frame. 4. Main frame. 5. Guide-yoke. 6. Guide-bars.

boiler-support, which are clearly indicated, we have the framework belonging to the engine proper completed by the guide-yoke 5 and the guide-bars 6; the latter being carried by the cylinder-head at their front end, as can be seen also in Figs. 247 and 256. The small bearing which rests upon the frame just in front of the guide-yoke is for the rocker-arm, a part of the valve-gear.

### § 43. The Cylinder.

(a) THE CYLINDER OF A SIMPLE LOCOMOTIVE of moderate size is very fully illustrated by the sectional views in Figs. 247 and 248. There are several reasons why this particular type is chosen for detailed description: its general form is simple, even though some complication is introduced by the connection to the body of the saddle-casting; in the main view it is almost entirely clear of the framework; and especially, as regards the short slide-valve and the long steam-passages, it is the extreme type in one direction, with the cylinder having separate valves and short ports as the other extreme.

The cylinder-body 1 is a continuous shell, except where the steam-ports 9, 9, cut through it; and around each end there is a stout flange to which the cylinder-heads 3 and 4 are fastened with stud-bolts. That part of the inside surface along which the piston slides is called the "bore" of the cylinder: at the ends it is counter-bored a little larger (from  $\frac{1}{8}$ " to  $\frac{1}{4}$ " in diameter). This is done partly to facilitate re-boring when worn, partly in order that the piston may not wear the rubbing surface to a shoulder at the end of the stroke—the edge of the packing-ring slightly over-travelling the end of the bore. The heads, shallower than in any of the designs which follow, extend into the counterbore, and are stiffened with radial ribs on the outside. The back head carries the stuffing-box—of special form in that it is to contain metallic packing—and a stout flange for supporting the guide-bars, as already shown on Fig. 246. The piston is of simple cast-iron box form, and calls for no comment at this point.

(b) DETAIL OF THE CASTING.—Turning from the cylinder proper to the valve-chest and steam-passages, and to the portion

of the casting which joins the cylinder to the saddle, we encounter much more complicated shapes. This rather intricate part of the casting will now be fully described, not only on account of importance as a type of design, but also because its very complex

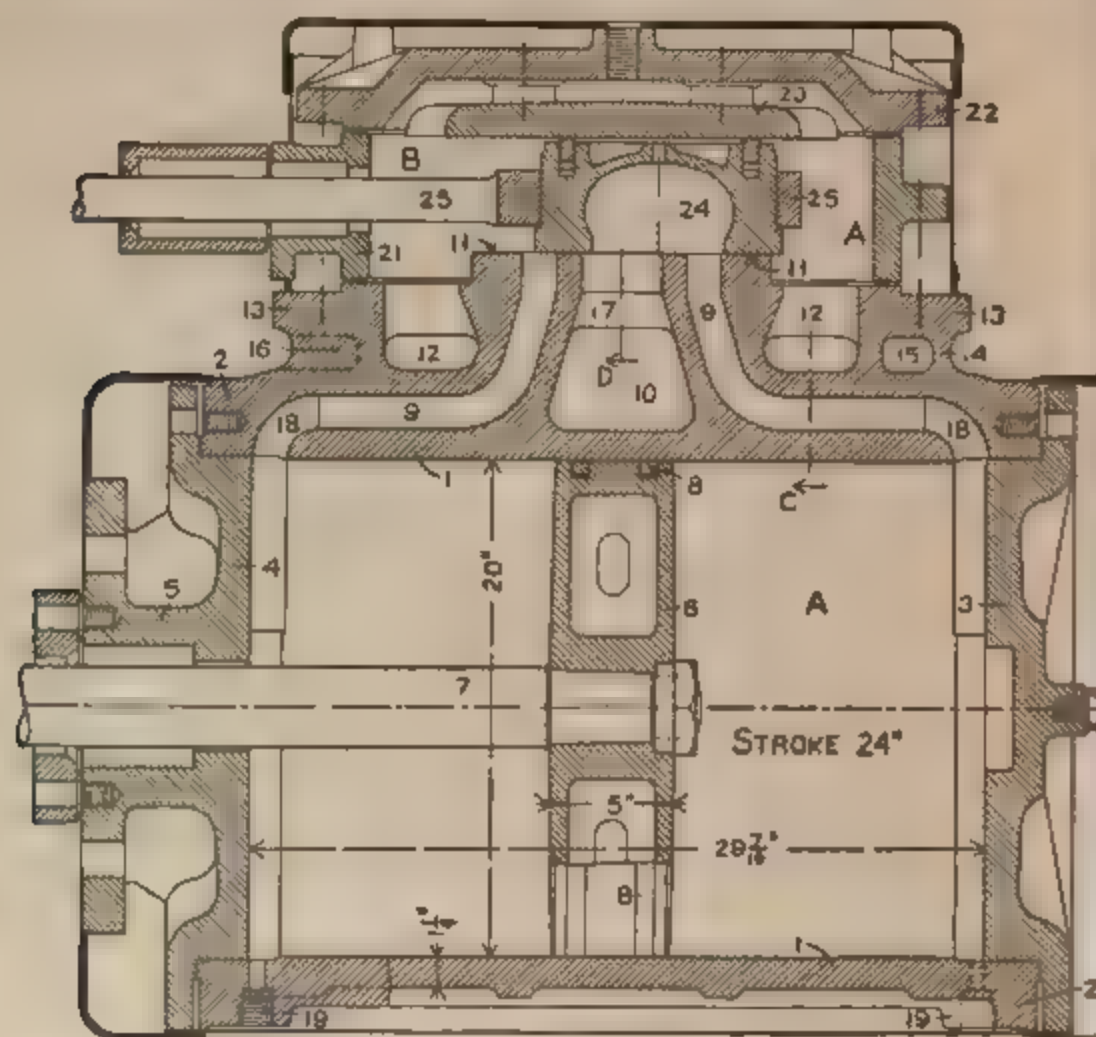


FIG. 247.—Lengthwise Section of a Locomotive Cylinder, 20" diameter by 24" stroke. Scale 1 to 12.

1 to 8. Main Outline of Cylinder.

- |                             |                   |
|-----------------------------|-------------------|
| 1. Cylinder body or barrel. | 5. Stuffing-box.  |
| 2. Cylinder flanges.        | 6. Piston.        |
| 3. Front cylinder-head.     | 7. Piston-rod.    |
| 4. Back cylinder-head.      | 8. Packing-rings. |

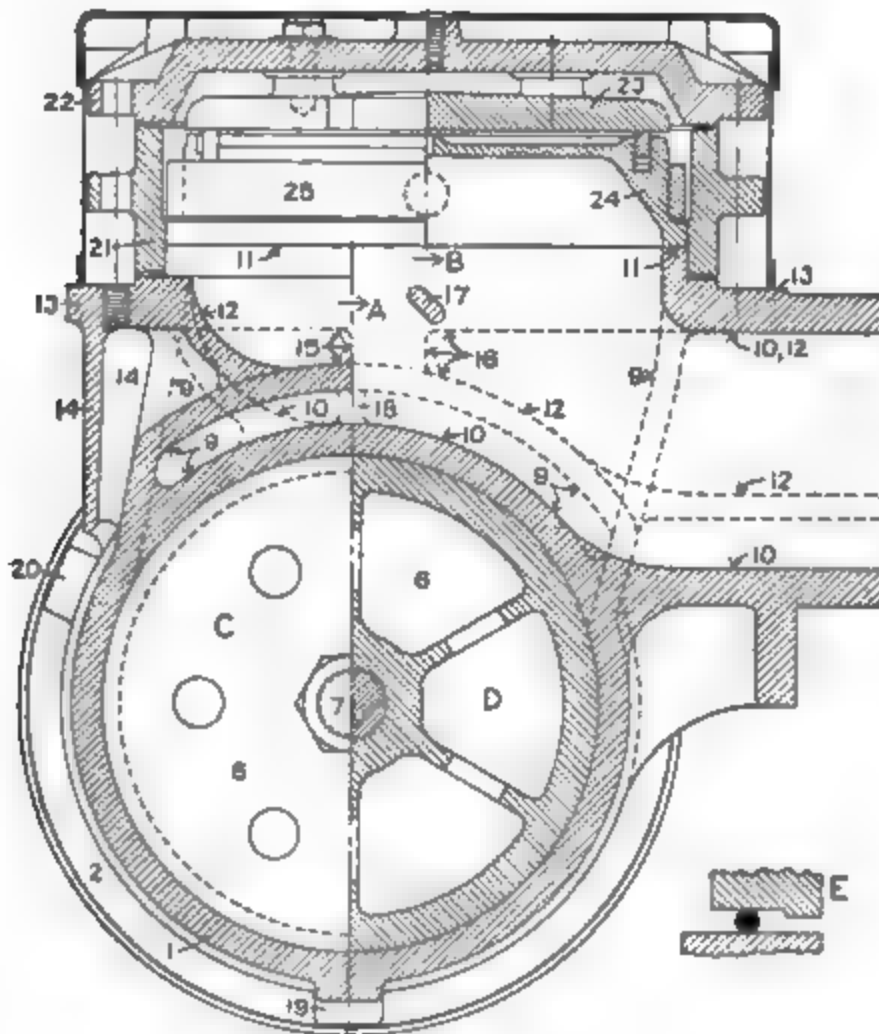
9 to 13. Steam passages.

- |                   |                      |                  |
|-------------------|----------------------|------------------|
| 9. Steam-ports.   | 11. Valve-seat.      | 12. Steam-inlet. |
| 10. Exhaust-port. | 13. Valve-chest box. |                  |

entails some excellent practice in reading drawings and in appreciating the essential form of a casting which has to fulfil certain requirements.

The bottom of the steam-chest is a flat, rectangular to

through which open the steam-passages; and in the space between this table and the cylinder-shell are formed the walls of these passages. The steam-ports 9, 9, and the exhaust-port 10 come up to the raised valve-seat 11. The steam-pipe, coming down



**FIG 248.—Cross-section of Cylinder in Fig. 247.**

**14 to 20. Details of Cylinder-casting.**

- 14. Apron.
- 15, 16. Hollows around steam-passages.
- 17, 18. Stiffening-struts across ports.
- 19. Drip-cock taps.
- 20. Indicator taps.

**21 to 25. Valve-chest and valve.**

- 21. Valve-chest casting.
- 22. Valve-chest cover.
- 23. Balance-plate.
- 24. Valve.
- 25. Valve-yoke and rod.

through the saddle (see Fig. 246 and Fig. 257), forks into the two steam-inlets 12, 12, which open into the steam-chest at each end of the valve-seat. In Fig. 248 the profiles of the different ports are indicated by repeating these numbers, and marking with arrow-heads the lines to which they refer. The steam-

ports drop down between slanting plane walls, then curve around and are fitted to the outer surface of the cylinder, until they go to the ends and turn in—being of nearly constant width. The exhaust-port is bounded by easy curves as it drops down to the surface of the cylinder, then runs off into the saddle; and the steam-inlets are of similar shape, but are above the outer wall of the steam-ports.

To keep the metal around these various cavities uniform thickness, a number of pockets must be formed in the external surface of the casting. Inside the apron 14, which at the ends bends around to join the cylinder-flanges, there is, in the middle, a pocket coming to the wall of the exhaust-port 10; then the steam-port walls, coming down straight, and connected to the apron by ribs which stiffen the latter; next, shallow pockets in the steam-inlet walls; and last, at the ends, deep pockets which run in as far as the central plane of the cylinder. The minimum section of the space 15 is shown on Fig. 247, at the right; at the left is given the outline of a similar space 16 at the back of the cylinder, but which is not closed in like 15. To stiffen the port-walls, the stud 17 is cast in the exhaust-port, set at an angle so as to obstruct the passage as little as possible; and similar cross-studs 18, 18, tie the flanges to the cylinder-body near the middle of each steam-port opening.

Other details, external, are, the stout rib along the bottom of the cylinder; the enlargement of the outer diameter at the counterbores, with two shallow ribs running around the cylinder and the taps for the cylinder-drains and the indicator pipes, 19 and 20. The form of the saddle-casting just back of the cylinder is sufficiently shown on Fig. 246.

(c) VALVE-CHEST AND VALVE.—The steam-chest 21 is a rectangular box, with a stiffening-rib all around it and the valve-restuffing-box at one end. The cover 22 is a separate piece, and one set of stud-bolts tightens up both joints. The device for making these steam-tight is shown in detail at E, Fig. 248; a rectangular "ring" of heavy copper wire, soldered at the corners, is placed between the cast-iron surfaces; when the bolts are screwed up the wire is squeezed flat, and the ledge on the steam

chest keeps this gasket from blowing out. The same kind of packing is used at the stuffing-box covers. The cylinder-heads, however, have "ground" joints, and need no packing.

Beneath the steam-chest cover is bolted the balance-plate 23, whose lower surface is a true plane, parallel to the valve-seat. The valve is a simple D valve, short and wide, and the valve-yoke 25 goes clear around it, as shown by the top view in Fig. 249. This view also makes clear the manner in which the

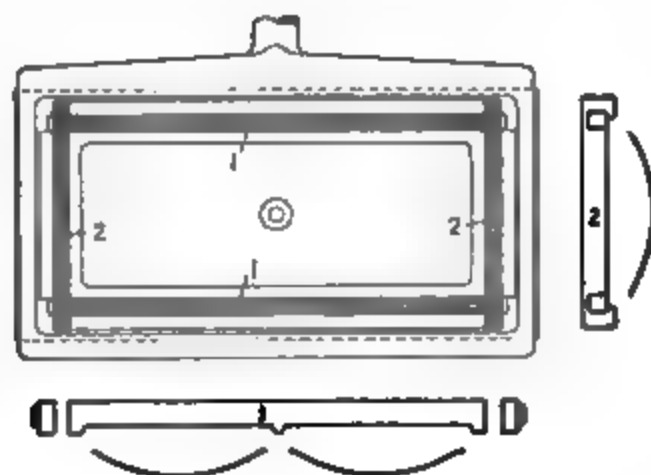


FIG. 249.—Plan of Valve and Valve-yoke in preceding figures.

balance-strips, shaded for emphasis, shut off steam from the greater part of the top of the valve. In this design—the Richardson balanced valve—there are four straight strips, fitted neatly into the grooves in the valve, and touching each other closely at the corners; the end strips 2 having lugs to keep them in place. Under the strips are light springs; but they are pressed up against the balance-plate chiefly by the steam, which has free access to their under side. Note the small vent-hole, connecting this relieved space with the exhaust-port.

(d) VARIOUS ENGINE-CYLINDERS.—The drawings which follow, Figs. 250 to 273, show a number of different steam-cylinders, more or less in detail, for the purpose of bringing out the important variations in arrangement and construction. In first running over these, we shall note the more striking features of each design; then, after all the examples have been presented, some of the more important details will be systematically reviewed. It is self-suggestive that there are two principal points of view from which



the cylinder may be regarded: on one side is the function of work performance, involving the action of forces, and presenting questions of strength, tightness, durability, and convenience of access; on the other side is the function of steam-distribution. The working element, the "cylinder" proper, shows only secondary variations in the form of its parts, however important these differences may be from the point of view of the constructor: but in the form and arrangement of the valve (or valves) and of the steam-passages there is a wide variety. Note that of the two elements of the steam distribution, form and movement, we are now concerned with the first only, except in the most general way; and the various valves shown can be more fully understood, in their details and proportions, after the action of the valve-gear has been studied in the next chapter.

(e) DIFFERENT VALVE-ARRANGEMENTS.—Analyzing the steam distribution, as already described in Chapter I., we note that a valve in an ordinary double-acting engine has to perform four functions, namely, to admit steam to each end of the cylinder and permit exhaust from each end. The common D valve, with each of its four edges controlling one of these operations, may well be called a four-function valve. In many cases, the functions are separated, two double-function valves, or four single-function valves, being used. Of the latter arrangement, the Corliss engine is the most prominent example.

(f) ENGINES WITH SEPARATE SLIDE-VALVES.—The example given in Fig. 250 is typical of a number of designs. With its four short ports and separate valves, it contrasts very strongly with the locomotive cylinder in Fig. 247. Structurally, this engine has four valves; kinematically—that is, as regards movement—there are two valves, one for the two admissions, the other for the two exhausts. Right here a distinction can be drawn between separate valves and a separated valve. When a plain or parallel-balanced D valve is used, as in Fig. 247, it is made as short as possible, in order that there shall be the minimum of area upon which the steam-pressure may act to force the valve hard down upon its seat and produce friction. When the valve is fully balanced, as in the Sweet design, shown in Fig. 6, it is made long



or the working-faces are separated, so as to get short ports. Complete evolution in this direction is shown in Fig. 250, where the valve is divided into two separate parts, connected only by the valve-rod which moves them. If now these two valves are separately driven, as in the Porter-Allen engine, we have what may strictly be called separate valves. The Porter-Allen, one of the earliest-developed good high-speed engines, is not here illustrated; but the valve-gear is shown in outline in § 58, and the general arrangement of the cylinder is very much as in Fig. 250.

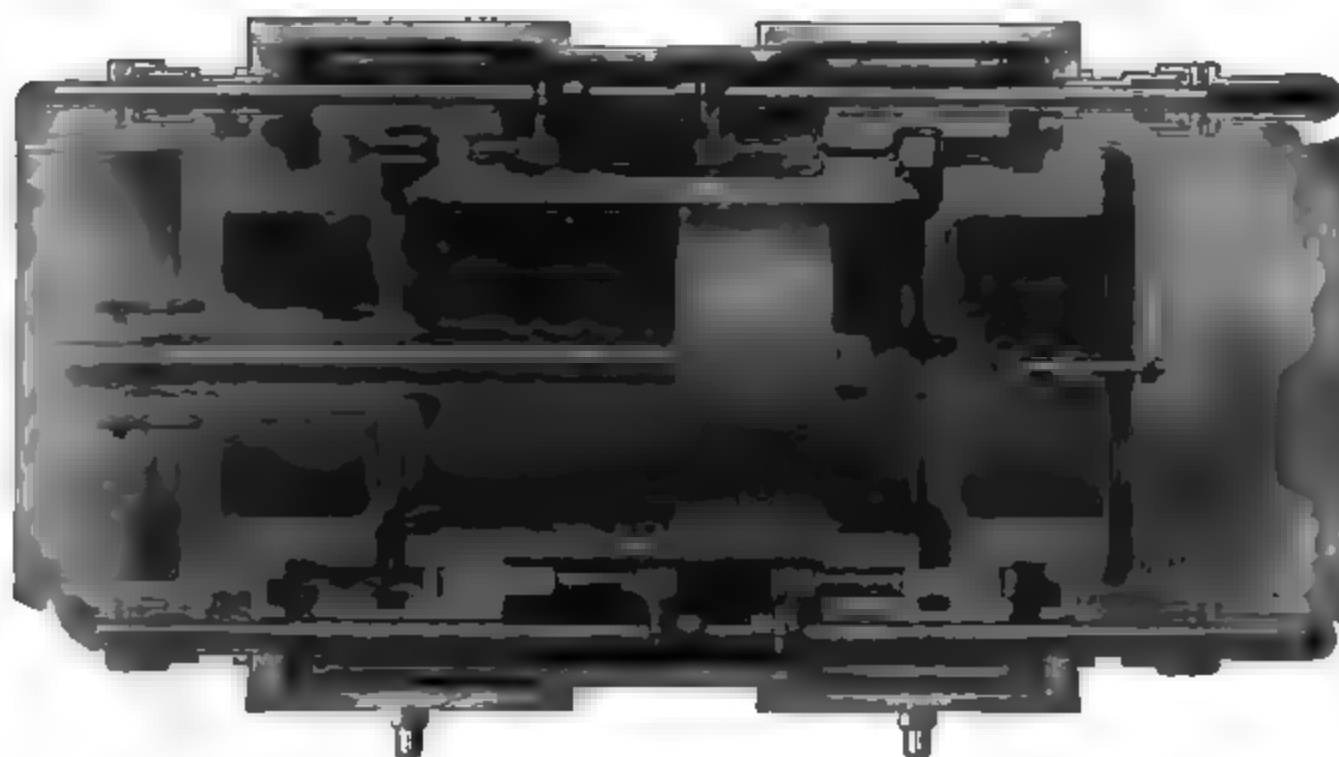


FIG. 250.—Cylinder of Watertown Four-valve Engine.

A noticeable feature of this cylinder is the great depth of the heads, which is rendered necessary by the requirement of the steam-chest space well beyond the ports. In regard to the balancing of the valves, we see that the balance-plates on the exhaust side must be held in place mechanically, against the distance-strips which just leave the valve room to move; whereas the balance-plates on the admission-valves are held up by the steam-pressure, like the single plate in Fig. 6. Further, each exhaust-valve has two or three slots cut through it (rather than the single one shown in this illustration), in order that the steam-pressure shall always be equalized on the two sides of the valve,

or that the pressure of the steam in the cylinder shall not force the valve hard against the balance-plate.

(g) THE DOUBLE-FACED FLAT VALVE.—The cylinder of the engine illustrated in Figs. 201 and 237 is shown in detail by Fig. 251, with especial emphasis on the valve and steam-ports. In

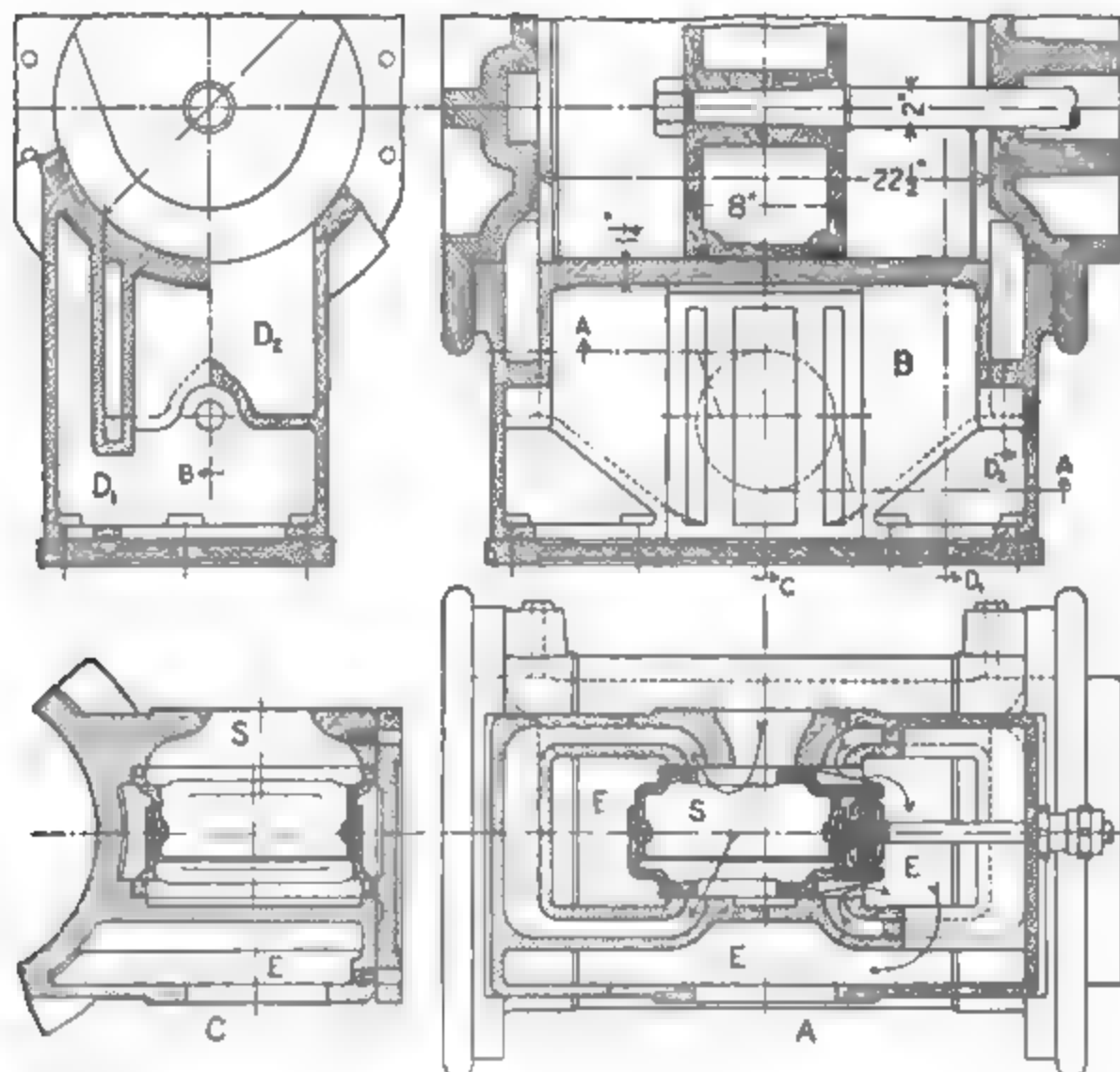


FIG. 251.—Cylinder of 18" by 14" American-Ball Engine; see Figs. 201 and 237. Scale 1 to 14.

this Ball design there are two parallel valve-seats, just alike, with the extensible valve between them. Each half of the valve has the usual working-face of a common D valve; and the halves are connected by a sliding cylindrical joint, telescopic in its action and with grooves and a packing-ring to prevent leakage. This valve differs from those in Figs. 6 and 247 in that the live steam

is at the middle, while the exhaust takes place past the outer edges of the valve. This interchange of function does not affect the steam distribution at all; its effect upon the arrangement of the valve-gear will be brought out in the next chapter, where the "balancing" of the valve will also be discussed. Note how the valve-rod is connected to the valve, by a self-adjusting grooved block, free to take the position in which there will be no bending effect upon the rod.

This horizontal position of the valve, in a plane through the cylinder-axis, makes the ports long, so that the engine is bound to have a large clearance-volume: but the mere doubling of the

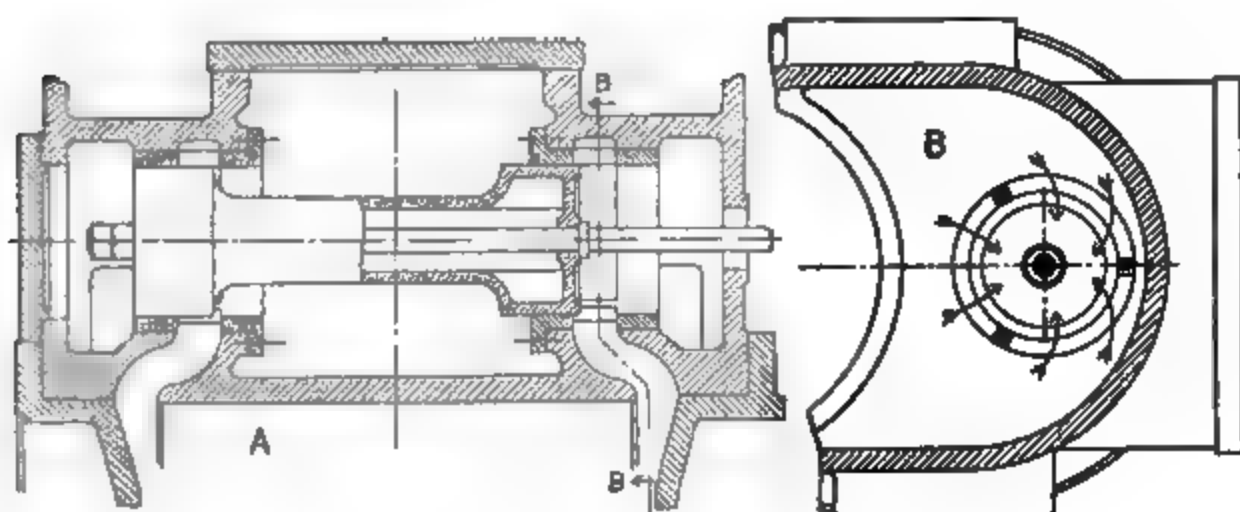


FIG. 252.—Cylinder with Plain Piston-valve. Scale 1 to 12.

ports does not of necessity give them a larger total cross-section than a single passage ought to have. The shape of this part of the casting is so fully shown by the sectional views as to need no explanation.

(h) ENGINES WITH PISTON-VALVES.—The first example, in Fig. 252, is of the simplest possible form—a plain, close-fitting plug, without any packing-device. This arrangement has been found quite satisfactory with valves not over six inches in diameter; but it is rather more usual to provide some means of adjustment or some sort of packing-rings even with small valves. The section along the port, at B, is useful in showing how the valve is placed near the back of the semicylindrical steam-chest, in order that

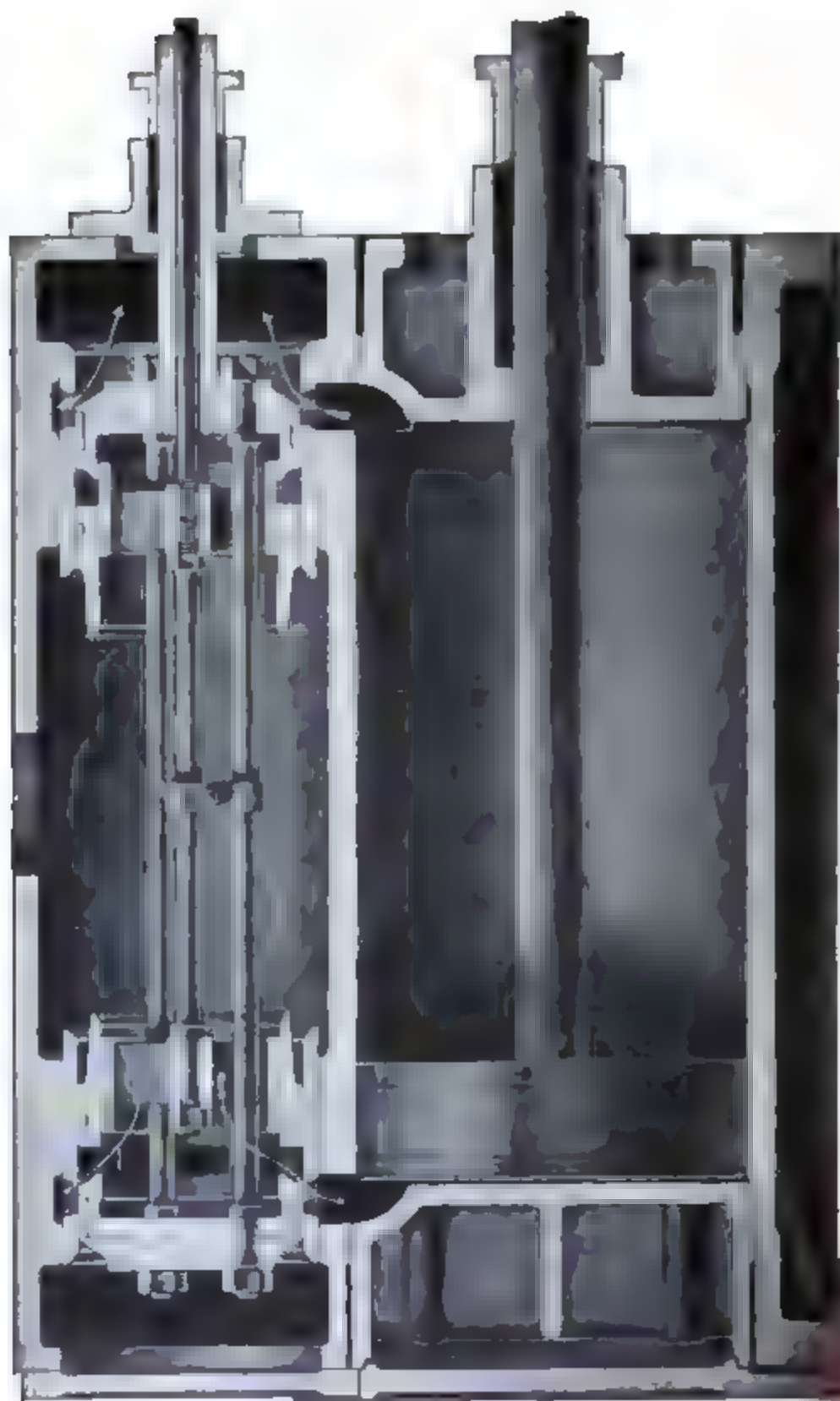


FIG 253.—Buckeye Cylinder, with Double Piston-valves.

the port shall fully and yet closely accommodate the streams flowing in or out all around the valve.

In Fig. 253 is seen a double piston-valve, each valve being in two parts, joined by rods. The inner, secondary valve combines with the outer main valve in controlling the operation of admission of steam into the cylinder; but the exhaust is determined wholly by the main valve. One valve-rod works inside the other, the hollow rod carrying a stuffing-box at its outer end.

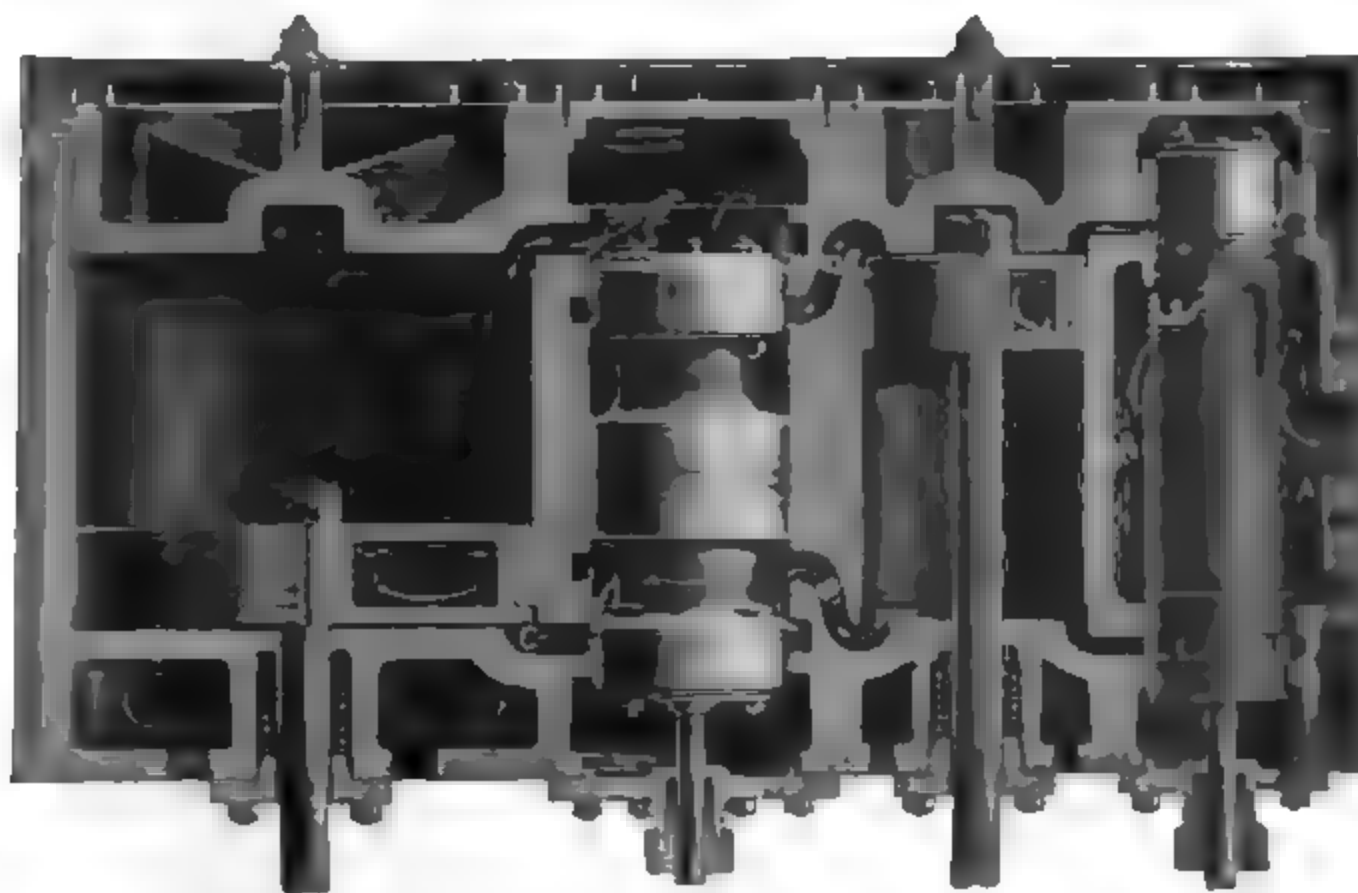


FIG. 254.—Cylinders of Reeves Vertical Compound Engine, Fig. 202

(i) **HIGH-SPEED COMPOUND ENGINES.**—In the very compact engine shown by Fig. 254, the two cylinders and their valve-chambers are cast all in one piece, with the lower cylinder-heads included. The working-faces of the piston-valves are adjustable in diameter. The high-pressure valve, at the right, works inside a full-length ring being completely inclosed except for the steam-inlets at the top; then the space AA forms a very effective steam-chamber within the engine, drained by the outlet at the bottom. The position for the particular position is indicated by arrows,

the working of the larger valve being somewhat complicated by the fact that it controls both the exhaust from the high-pressure cylinder and all the functions for the low-pressure.

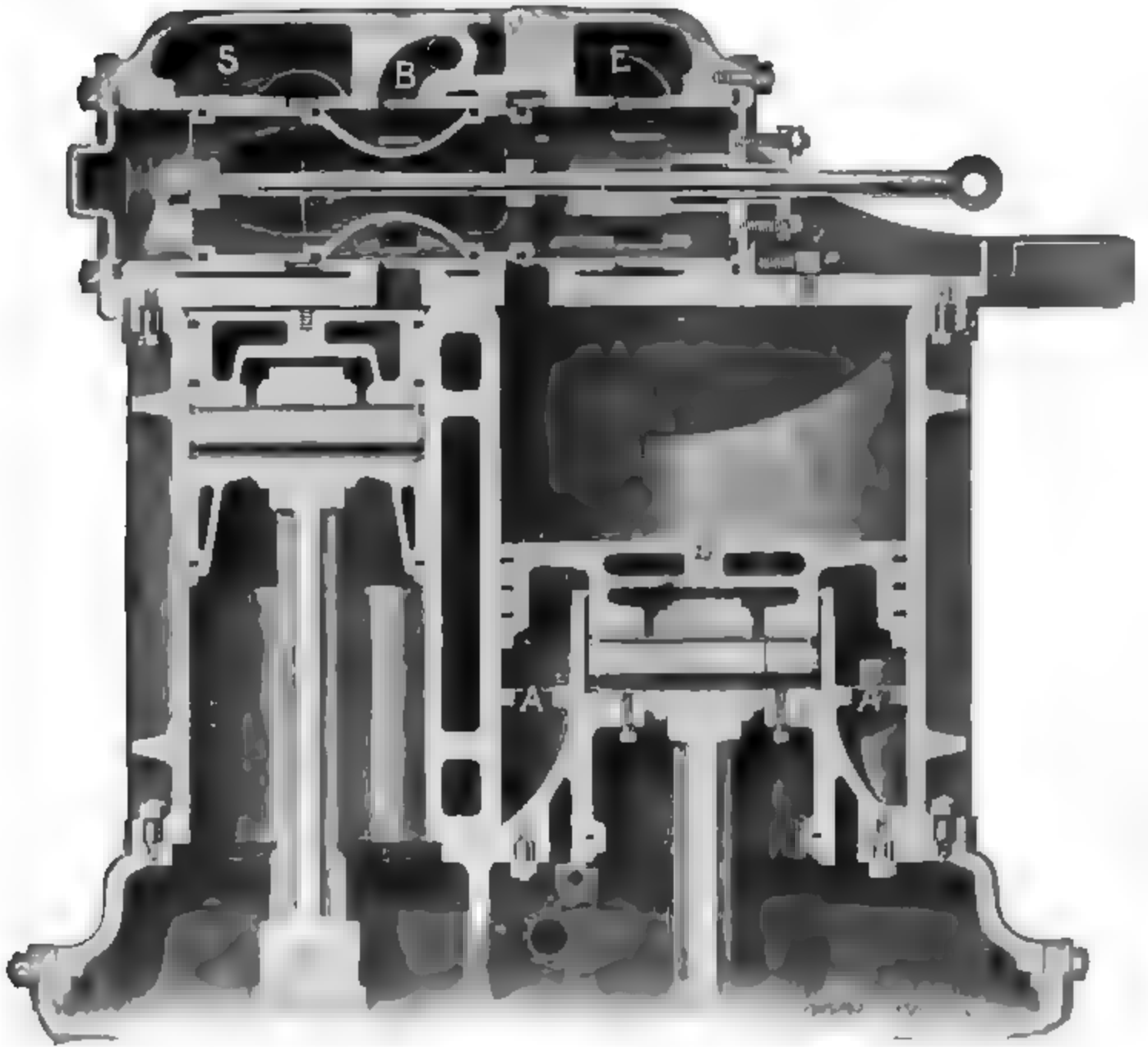


FIG. 235.—Cylinders of Westinghouse Compound Engine, Figs. 207, 208.

The general arrangement of the Westinghouse single-acting engine has already been commented upon under Fig. 208. In Fig. 225 are shown the details of the cylinders, pistons, and valve. Putting the steam-chest on top of the cylinders gives the latter

a very simple form. This sectional view makes clear the construction of the combined piston and cross-head. The external form of the valve is rather better shown in Fig. 208; here we see that it is hollow, all of its inner surface being exposed to the exhaust steam. The annular space around the neck of the valve alternately connects the steam-space S to the high-pressure port B, and this port to the low-pressure port; exhaust taking place past the right end of the valve, as in the figure. The valve works inside a continuous bushing, in which all the port-openings can be accurately machined. From B a by-pass valve connects to the steam-space

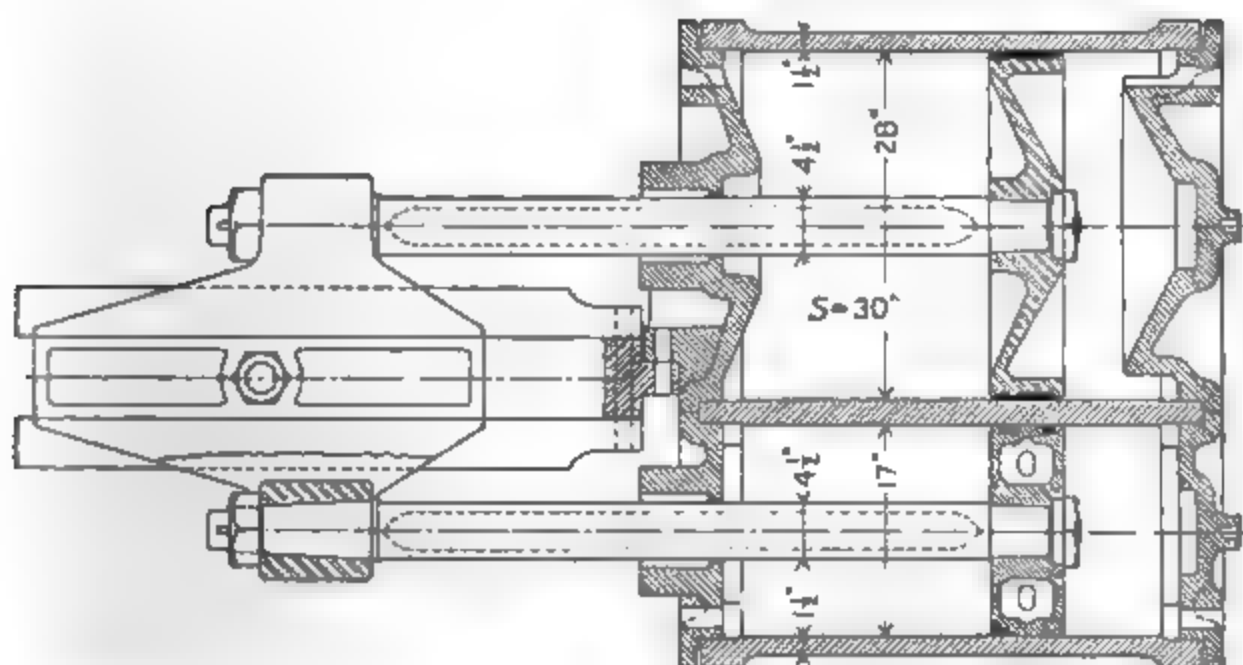


FIG 256.—Cylinders of Vaclain Compound Locomotive, Baldwin Locomotive Works. Scale 1 to 24.

S; this is worked by hand, and is opened when starting the engine, so as to admit steam directly to the low-pressure cylinder. From the annular space A, about the low-pressure piston-trunk, there is communication to the outer air through a check-valve, which can be held open while the engine is being turned over by hand.

(j) THE VAUCLAIN COMPOUND LOCOMOTIVE.—A particularly compact type of the direct-expansion compound engine is illustrated in Figs. 256 to 258. There are a number of interesting features, especially in the form and arrangement of the steam-passages and of the single valve which controls the steam-distribution of both cylinders. Fig. 256 is a vertical section, giving

the general arrangement of the cylinders, pistons, and cross-note the different kinds of piston used for the different diam. The details of the casting are shown by Fig. 257; while in Fig. 258 two partial sections, which follow the lines VH and VL in Fig. 257 A, are turned into one plane. According to the notation in the lengthwise sections, 1 or S is the steam-supply passage, the port to the high-pressure cylinder, and 3 to the low-pressure cylinder, and 4 or E is the exhaust-passage. On Fig. 257 A the p

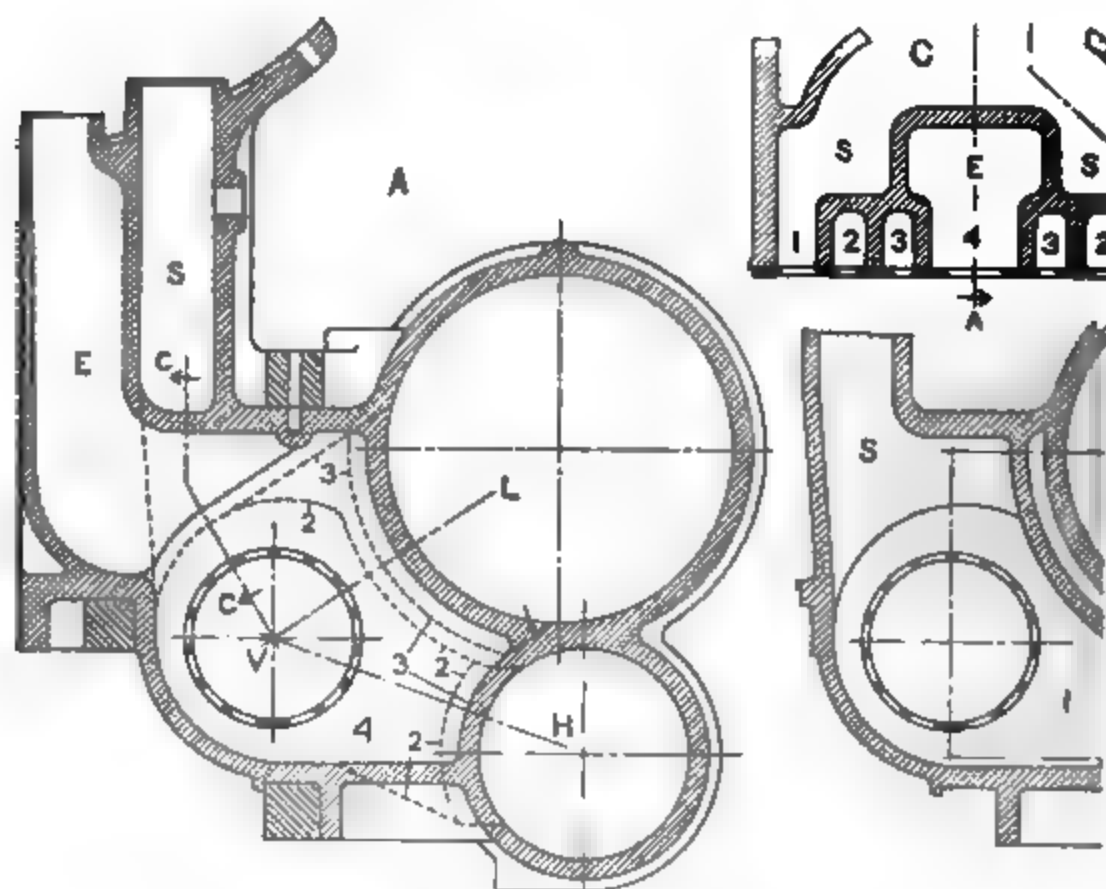


FIG. 257.—Sections of Cylinders and Saddle, Fig. 256. Scale 1 to

of the two ports are shown by dotted lines, each being marked with its number; with the help of Fig. 258 we see that 4 and 3 lead down to the walls of both cylinders, 2 is partly outside of the large cylinder, 1 is outside of both ports, 2 and 3.

In this particular example, the two cylinders have not their axes in the same vertical plane, as shown in Fig. 257 A; to diminish the over-all width of the locomotive, the large cylinder is moved out a little from the plane of the engine-mechanism, and the small cylinder is moved out a corresponding amount. Usually,



ever, the cylinders are placed one directly above the other; the large cylinder being on top in a freight locomotive with small wheels and a low stroke-line, the small cylinder on top when the wheels are large. An idea of the arrangement of the framework, at the saddle, can be got from view A.

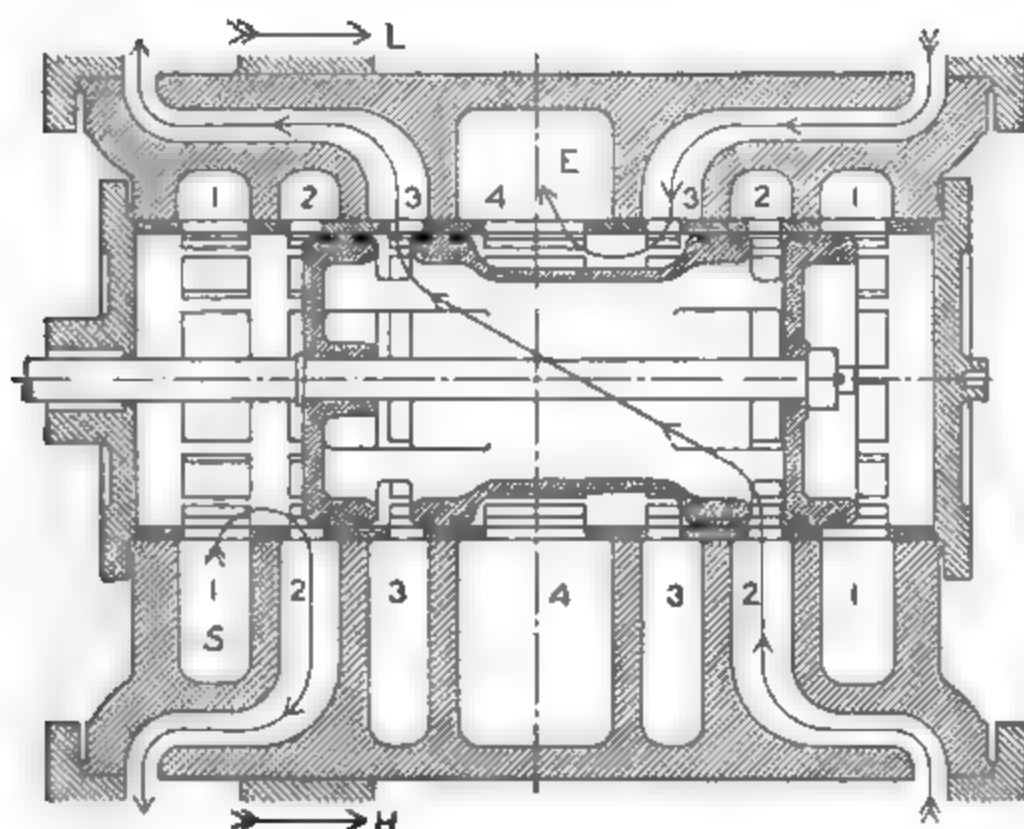


FIG. 258.—Sections through Valve and Ports, Fig. 256. Scale 1 to 18.

The form of the valve is clearly shown in Fig. 258, and the manner in which it controls the steam-action in both cylinders is indicated by the arrows. As with the D valve, the steam is here at the ends, the exhaust in the middle. A complete study of the action of this valve is quite complex—belonging to the subject-matter of Chapter XI. In the face of the saddle, on Fig. 257 A, is seen an outlet from the steam-passage; a by-pass valve connected to this can admit steam directly to the low-pressure cylinder, through the holes in the heads shown near the top of Fig. 256, in order to give extra power in starting the locomotive; but this steam is throttled down from the boiler-pressure, so that it will not exert too great a force upon the large piston.

Other cylinder-arrangements which are in the same class as the one just described are illustrated in Figs. 436 to 438.

(k) MARINE-ENGINE CYLINDERS.—As typical of this class cylinders of a small triple-expansion engine, already partly treated in Fig. 243, are fully shown in Figs. 259 and 260, and Fig. 411. Fig. 259 gives the general arrangement; the left half of the figure is a section at mid-length, the upper half through the top steam-port. The first point to be noted is the manner in which the cylinders are bolted together, so as to

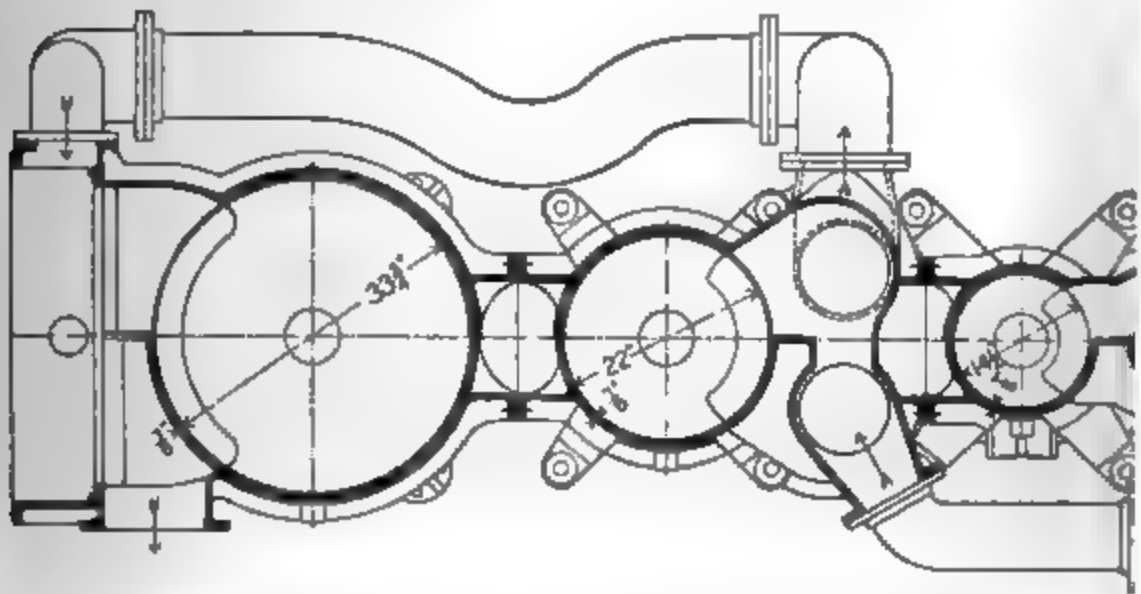


FIG. 259.—Cylinders of Small Triple-expansion Marine Engine; see Figs. 243, 260, and 411, stroke 18". Scale 1 to 32.

one rigid piece; besides the flanged seatings for this purpose see also the feet which rest upon the columns of the frame. As to the valve-arrangement, the use of piston-valves on high and intermediate cylinders and of a slide-valve on the low cylinder is very common; as is also the use of two or more piston-valves on one cylinder when it is large. Turning back to Fig. 243, we see that the H.P. valve receives steam at the middle and exhausts at the ends, through two outlets; these are joined by a branched pipe, which carries the steam to the middle of the intermediate valves; and a similar arrangement is used between the M.P. and L.P. cylinders. A good idea of the steam-pipe arrangement on marine engines can be got from Figs. 232 to 235;

ally the pipes are of copper; and expansion is allowed for by making the piping of a flexible form or by the use of slip-joints.

The detailed construction of two of the cylinders of the engine under consideration is given in Fig. 260. Cone-disk pistons, steel castings, are used; and the chief purpose in drawing parts of the two cylinders side by side is to show how the pistons have all

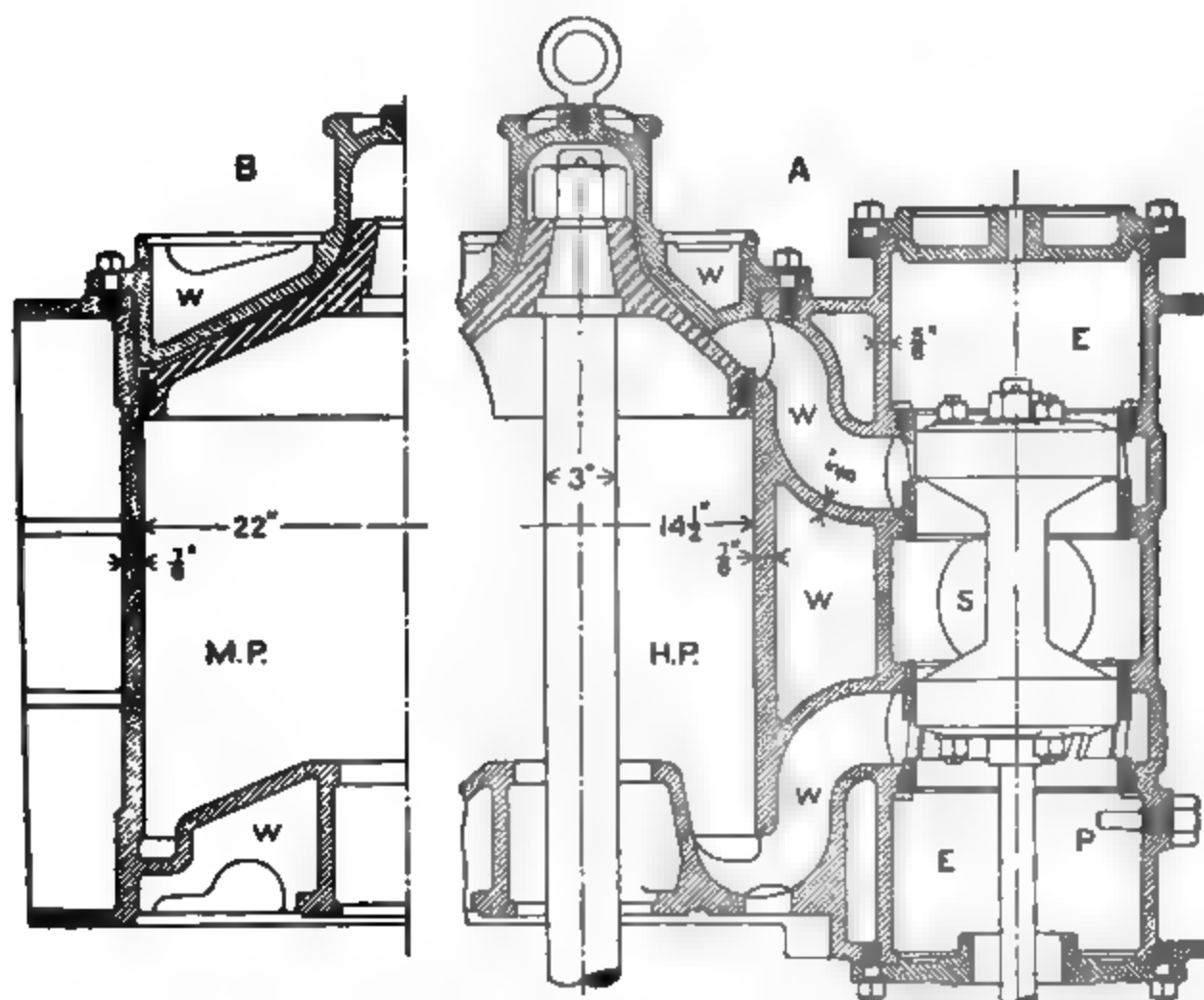


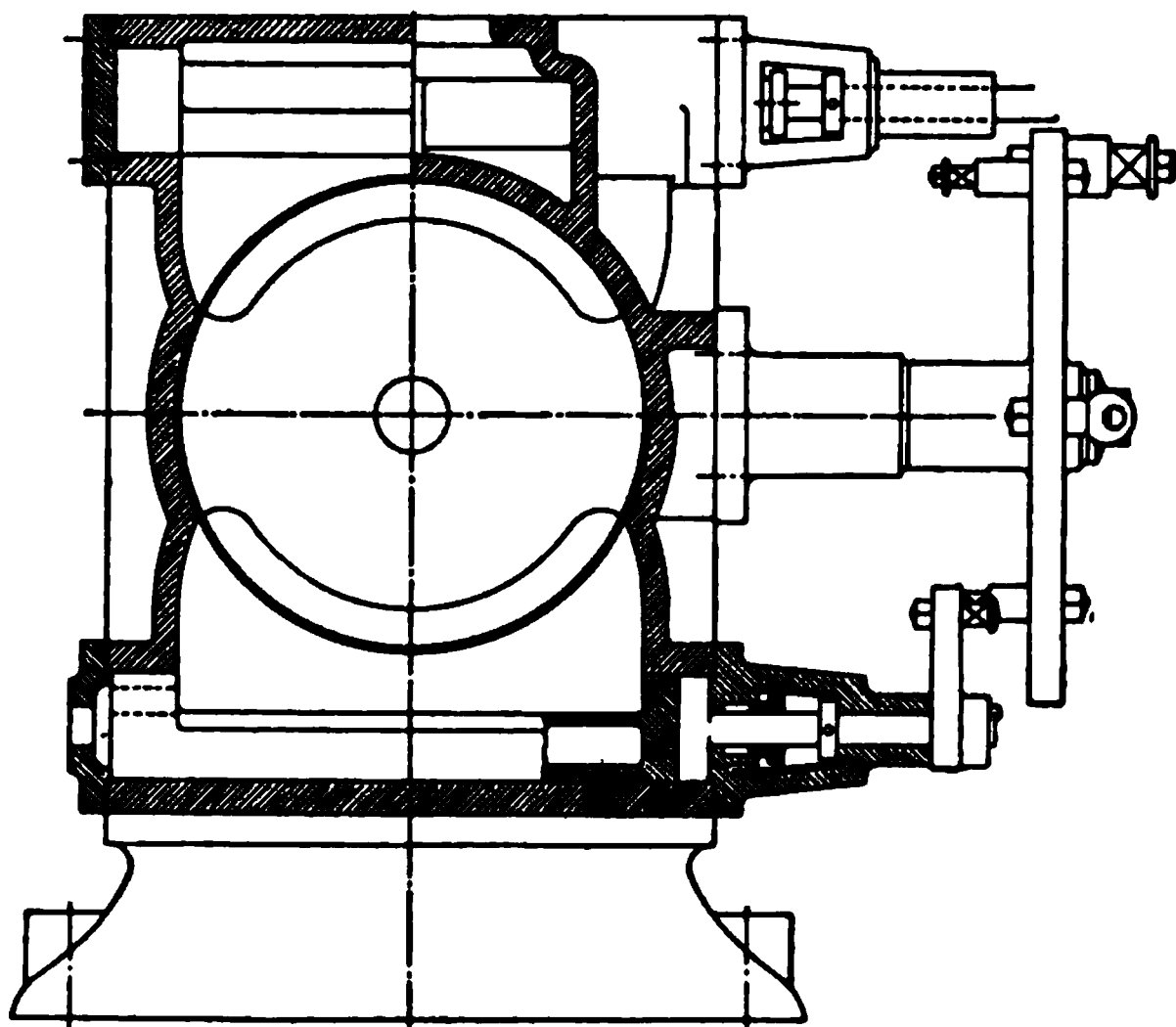
FIG. 260.—Details of Gunboat-engine Cylinders, Fig. 259 Scale 1 to 12.

the same height, so that the slant of the cone decreases with the diameter. The cylinder-heads are closely conformed to the piston, the lower head being cast solid with the cylinder and having an opening for the insertion of the stuffing-box. To separate the valve-chamber from the cylinder, uniting them by the ports and by stiffening-ribs as needed, is characteristic of marine practice. The two chambers on the intermediate cylinder are cross-connected in steam-space, ports, and exhaust-spaces. Note

the free use of ribs or webs (marked W) to stiffen the casting, especially in the ports.

A detail of this piston-valve, showing how it can be adjusted in diameter, will be found in Fig. 428. Other marine cylinders are more or less fully illustrated in Fig. 273 and Figs. 412 to 414.

(l) THE CORLISS CYLINDER.—A representative example of this type of cylinder is given in Figs. 261 and 262; of the former, three-fourths is a section through the head-end ports, while the upper



**FIG. 261.—Cross-section of Cylinder in Fig. 262.**

right-hand quarter is taken at mid-length. The general features calling for particular notice are, the location and form of the valves and ports, the steam-chamber at the top, and the exhaust-chamber at the bottom. Fig. 261 shows a part of the valve-gear, which will be found fully illustrated and discussed in § 56: for the present it is enough to know that the valves are oscillated back and forth so as properly to control the admission and exhaust for the two ends.

As to the valves, we note that each is a complete cylinder at the ends, with a full bearing in which it turns; but at its working part the steam-valve has only a narrow cylindrical face,

though the exhaust-valve fills more space. The valve-stem is a separate piece—the valves being, of course, made of cast iron—with a T head that fits into a slot cut in the end of the valve. The valve-chambers are closed at the ends by caps or bonnets, of which the one at the rear carries, on a yoke, the valve-stem bearing. In this engine a couple of large holes are drilled through the front heads of the exhaust-valves, to give passage to the relief-valves, which are screwed into the bonnets. On the back

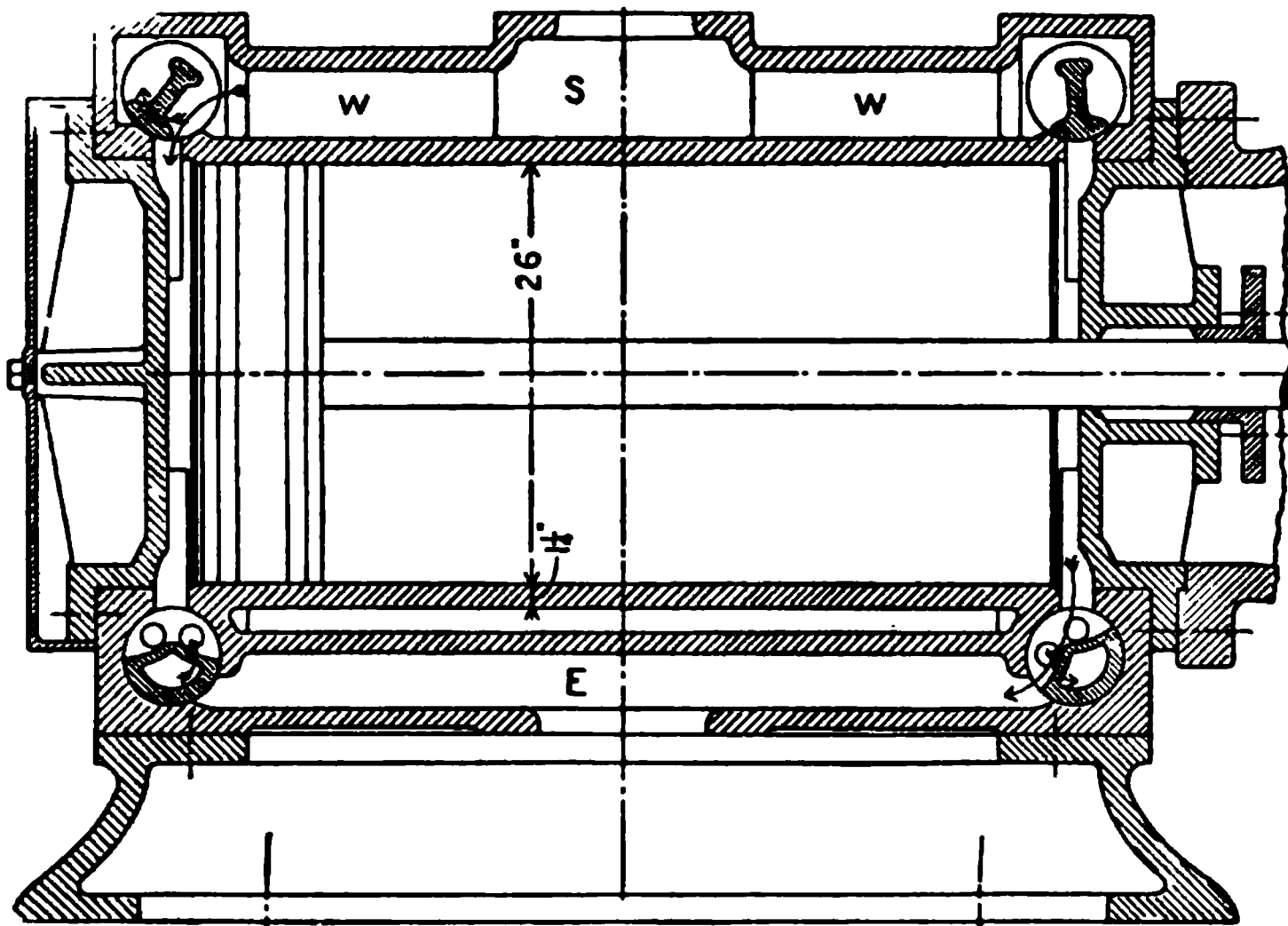


FIG. 262.—Lengthwise Section of Corliss Cylinder, 26" by 48".  
Scale 1 to 21.

of the cylinder, at mid-length, is formed a seating for the wrist-plate bracket.

The valve-chambers are here of the rectangular form, so that the cylinder has the square-cornered effect seen in Figs. 210 and 211. Sections of round valve-chambers are given in Figs. 241, 269, and 270: this closer conformation to the shape of the valve is usual in the later designs. In Fig. 262 the exposed surfaces of the valve-chambers are polished, in order to diminish the radiation of heat: with round-cornered cylinders, the sheathing covers everything. It is to be noted that the steam-chamber S is formed

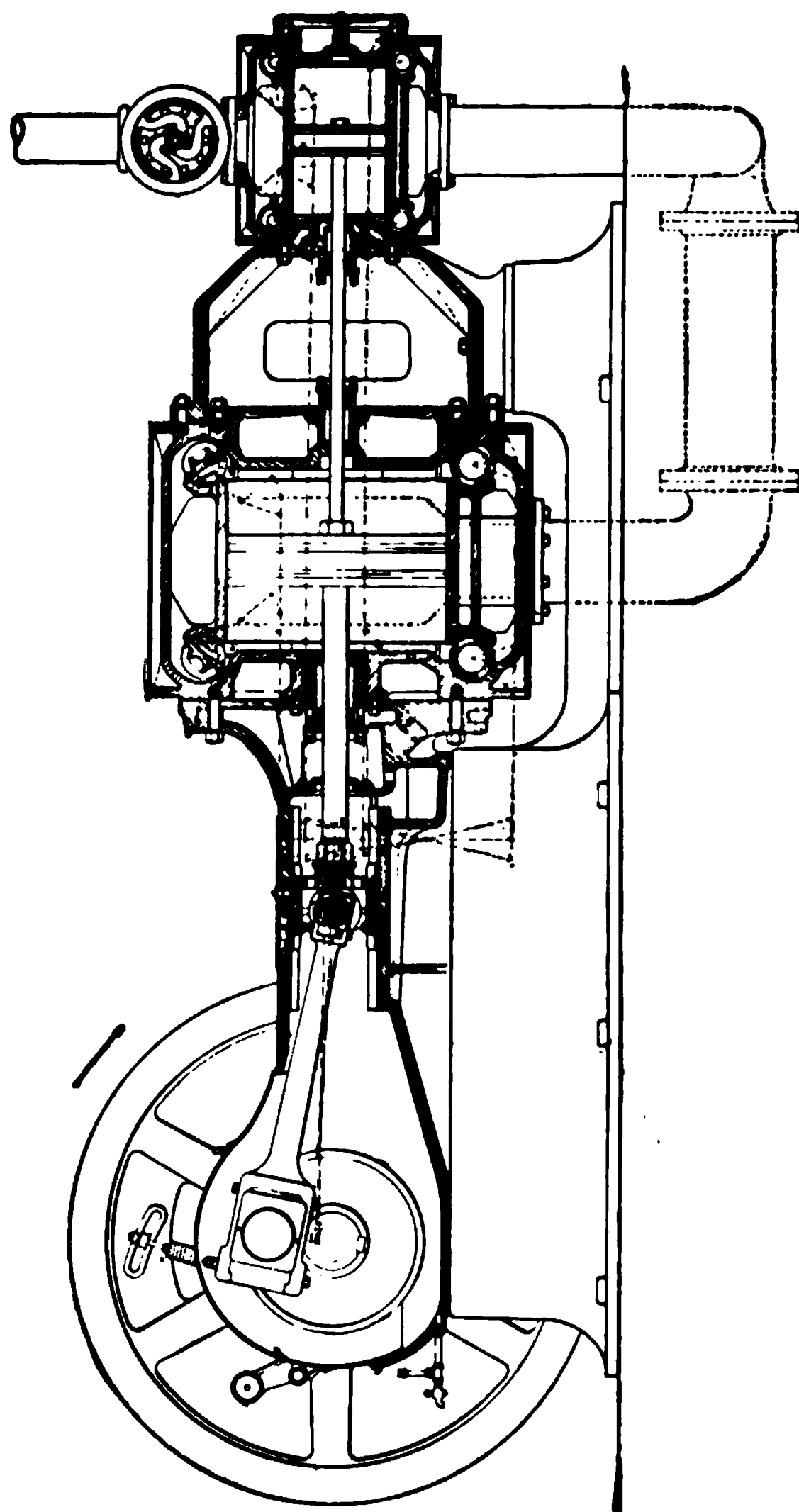


FIG. 263.—Section of Fleming Four-valve Tandem-compound Engine; compare Fig. 215.

right upon the cylinder, so as to steam-jacket a part of its surface; but the exhaust-chamber is separated from the cylinder-body. Stiffening webs, marked W, run along the middle of the steam-chamber. The main pipe-connections are obvious.

A high-speed engine with Corliss valves is illustrated in Fig. 263, which we now examine with especial regard to the form of the cylinder. Note the shield which extends beyond the valve-chambers, and forms the rectangular end better seen on Fig. 215. The valves of this engine are much more complex in shape than those in the preceding example, the steam-valves being triple-ported for small openings, double-ported for large ones. These, together with the valve-gear, will be more fully considered later.

(m) CYLINDERS WITH GRIDIRON VALVES.—After the Corliss type of oscillating plug valve, the next single-function type to be considered is the gridiron valve; which, by a simple sliding movement, opens and closes a number of ports. One prominent example is given in Figs. 264 and 265; the valves are long narrow slides with a number of openings running crosswise. In general arrangement this is very much like a Corliss cylinder, but with marked differences in the form of the valve-chambers. The valve-seats are separate inserted pieces, which is a decided advantage both as to original construction and as to ease of repair or renewal.

This engine has a positive valve-gear, of quite a complicated form, which will be fully discussed in the next chapter. To get a quick and variable cut-off, a second valve is used, working upon the back of each main steam-valve, and controlled by a shaft-governor, so that each cylinder has six valves in all. The exhaust-valves are closely guided between their seats, which are bolted fast, and the body of the cylinder: the steam-valves have more freedom; there is a clearance between the riding-valve and the ends of the fixed pins or studs that project inward from the valve-chest covers, so that the valves can rise about  $1/16$  inch from their seats; but they are kept from rattling, when running loose after steam is shut off, by the movable guides shown in Fig. 264. These are held by pins, which pass through an oblong hole; and each is pressed inward by a flat spring between its head and the cover.

The engine is shown in Fig. 264, which is a very clear diagram of the engine, showing the pressure in the cylinder, and the pressure in the water, which is accidentally

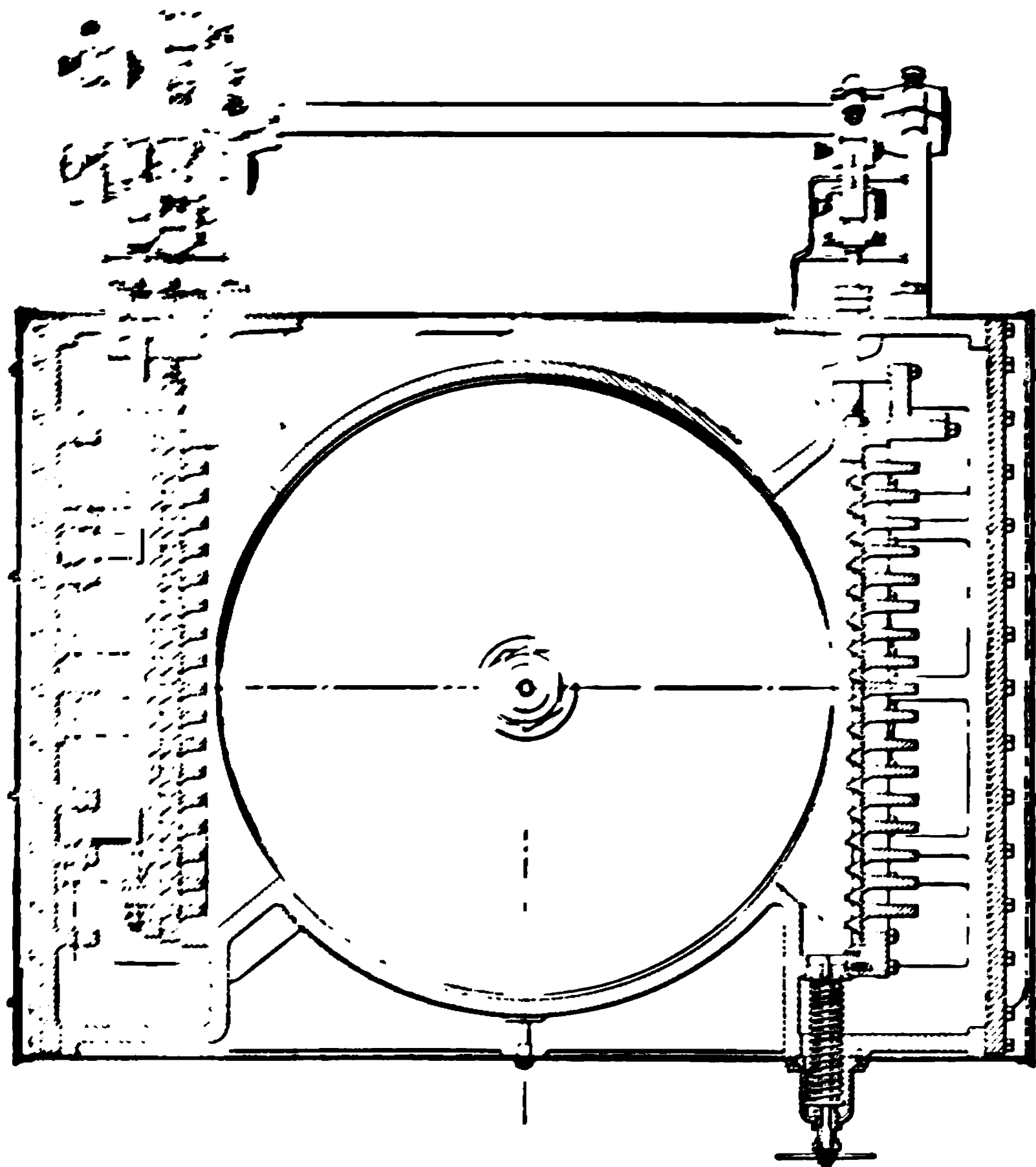


FIG. 264. Horizontal Section, low-pressure cylinder of vertical compound McIntosh & Seymour Engine; see Fig. 218.

drawn in. In general, a relief-valve is simply a spring-loaded safety valve; but the one here shown serves also as a drain-valve, since it can be opened by turning the star-wheel so as to draw



the valve back, compressing the spring. Note how this valve opens directly into the exhaust-chamber.

A very compact valve-arrangement, withal convenient of access, is that in the Wheelock engine, Fig. 266. The two grid-iron valves for one end are carried in a cylindrical plug, in which their seats are formed. This plug is inserted like a Corliss valve; it is given a slight taper, so that it will draw up tight; it carries the whole valve-gear on the head bolted to its outer end; and it can be removed when the valves are to be examined or repaired. This is the first steam-jacketed cylinder that has been shown: here the steam-chamber and the jacket are in all one, so that the

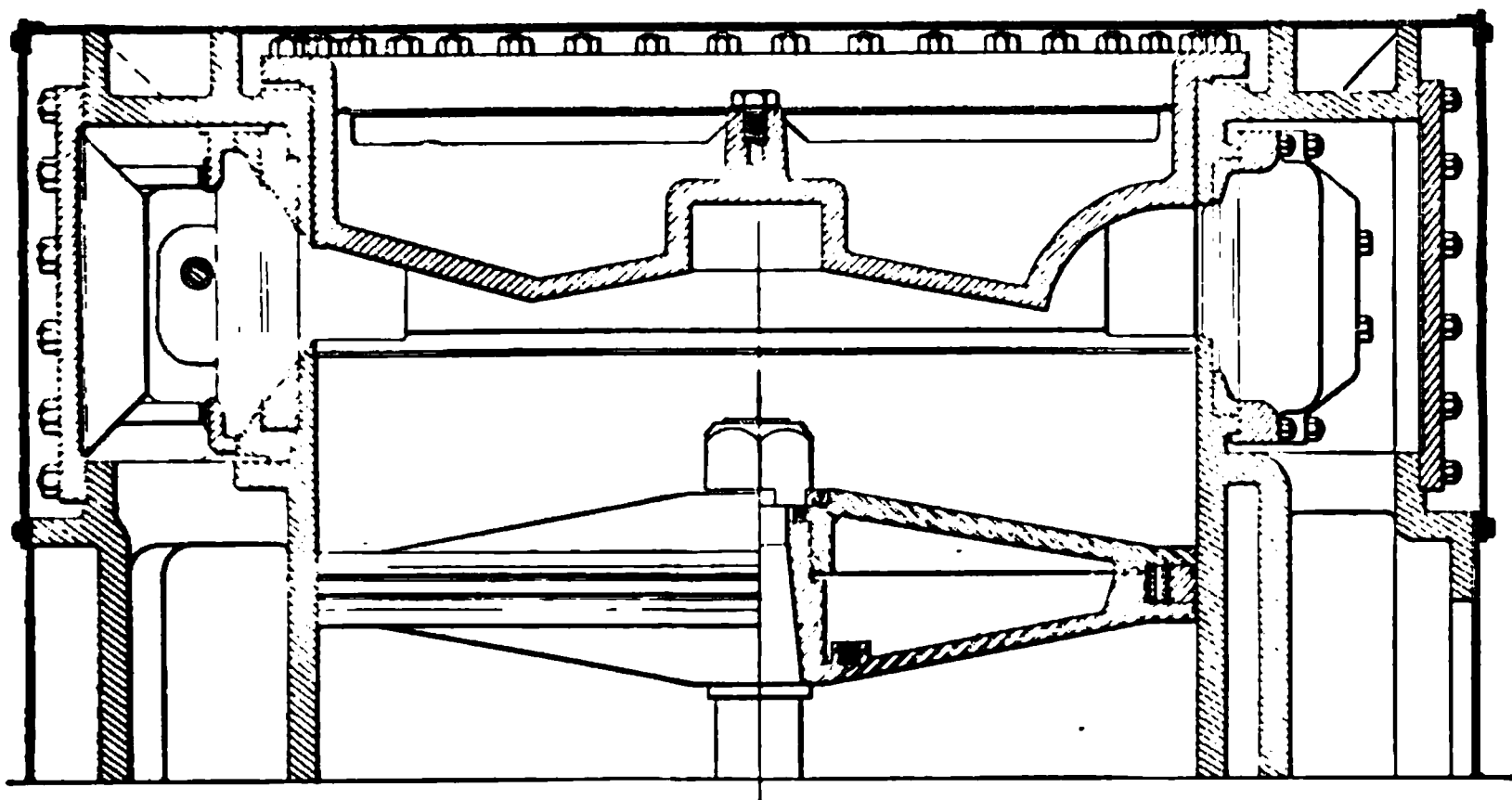


FIG. 265.—Axial Section of Cylinder in Fig. 264.

whole supply of steam passes through the jacket. The structural details involved in the arrangement will be touched upon presently; and illustrations of the valve-gear, which is of the releasing type, will be given in the next chapter.

(n) DROP-VALVE ENGINES.—All the valves so far shown act by the sliding of one surface upon another, whether the surfaces be flat or cylindrical. Radically different in form and action are valves of the lift or poppet type. These are not much used in American or English practice, and less now than formerly: but on the Continent of Europe, and especially in Germany, engines of the type illustrated in Fig. 267 are very common.

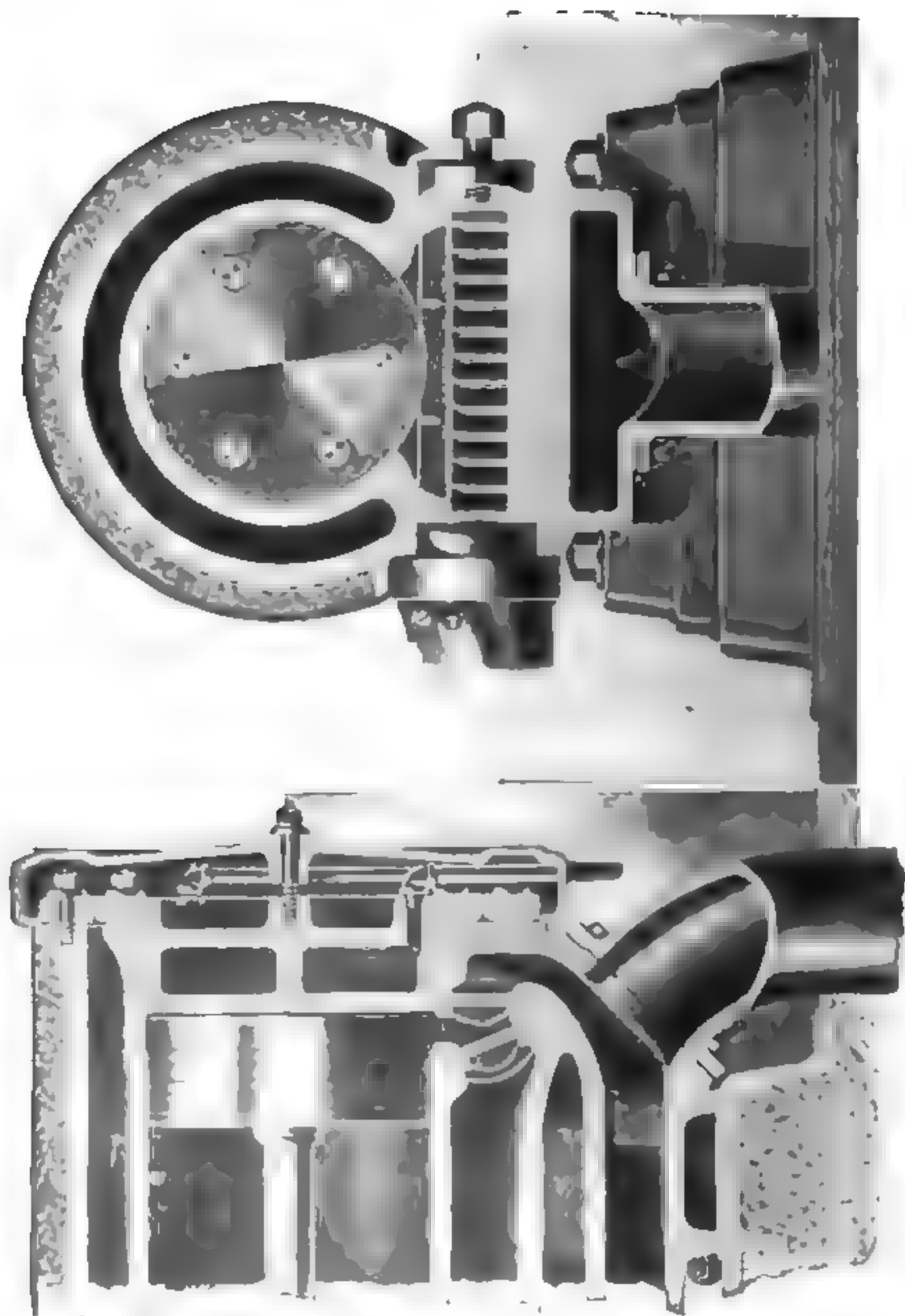


FIG. 206.—Cylinder of Wheelock Engine, FIG. 214.

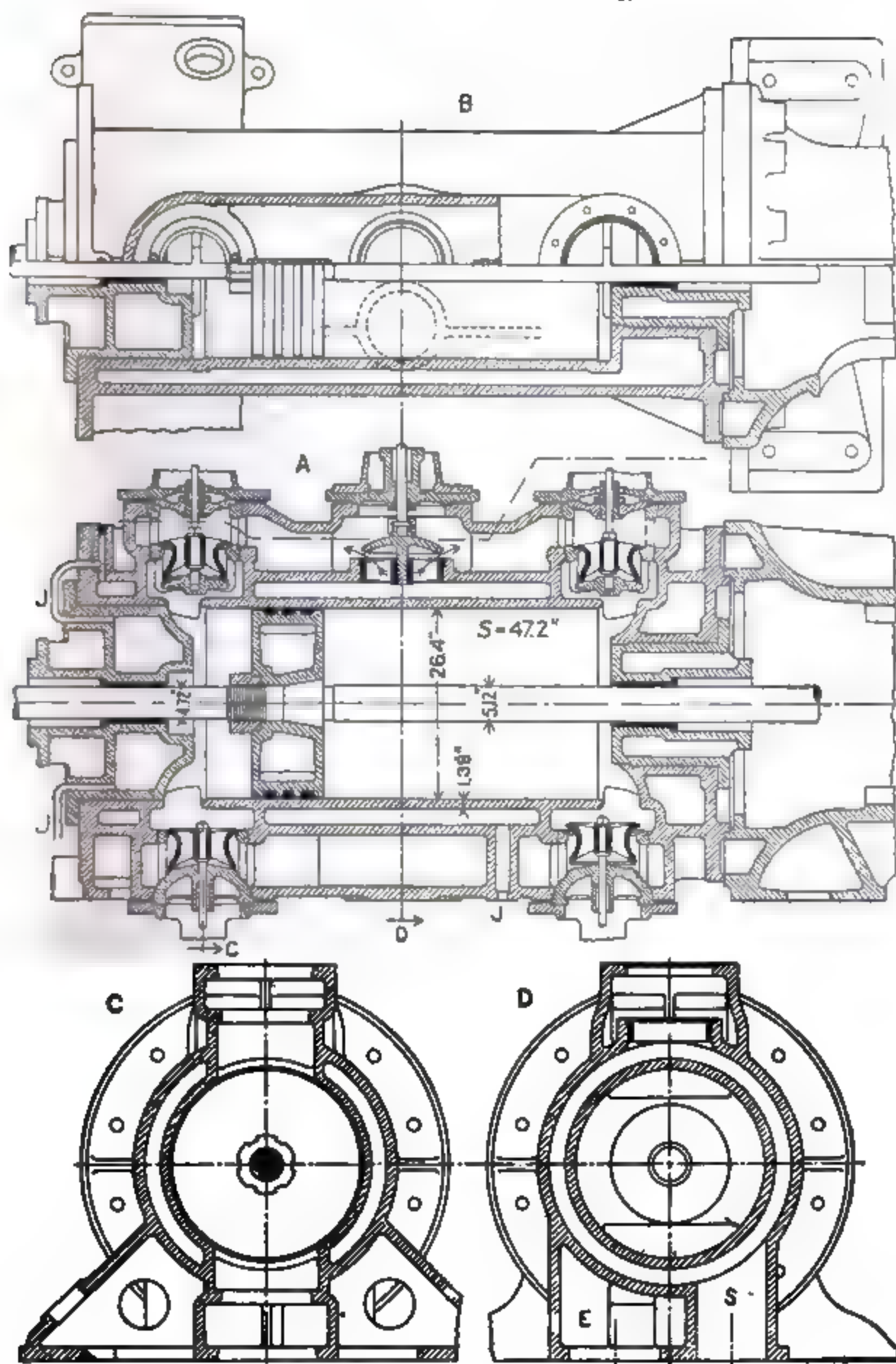


FIG. 267.—German Drop-valve Engine; high-pressure cylinder of 670 and 1075 by 1300 mm. compound engine. Scale about 1 to 80.

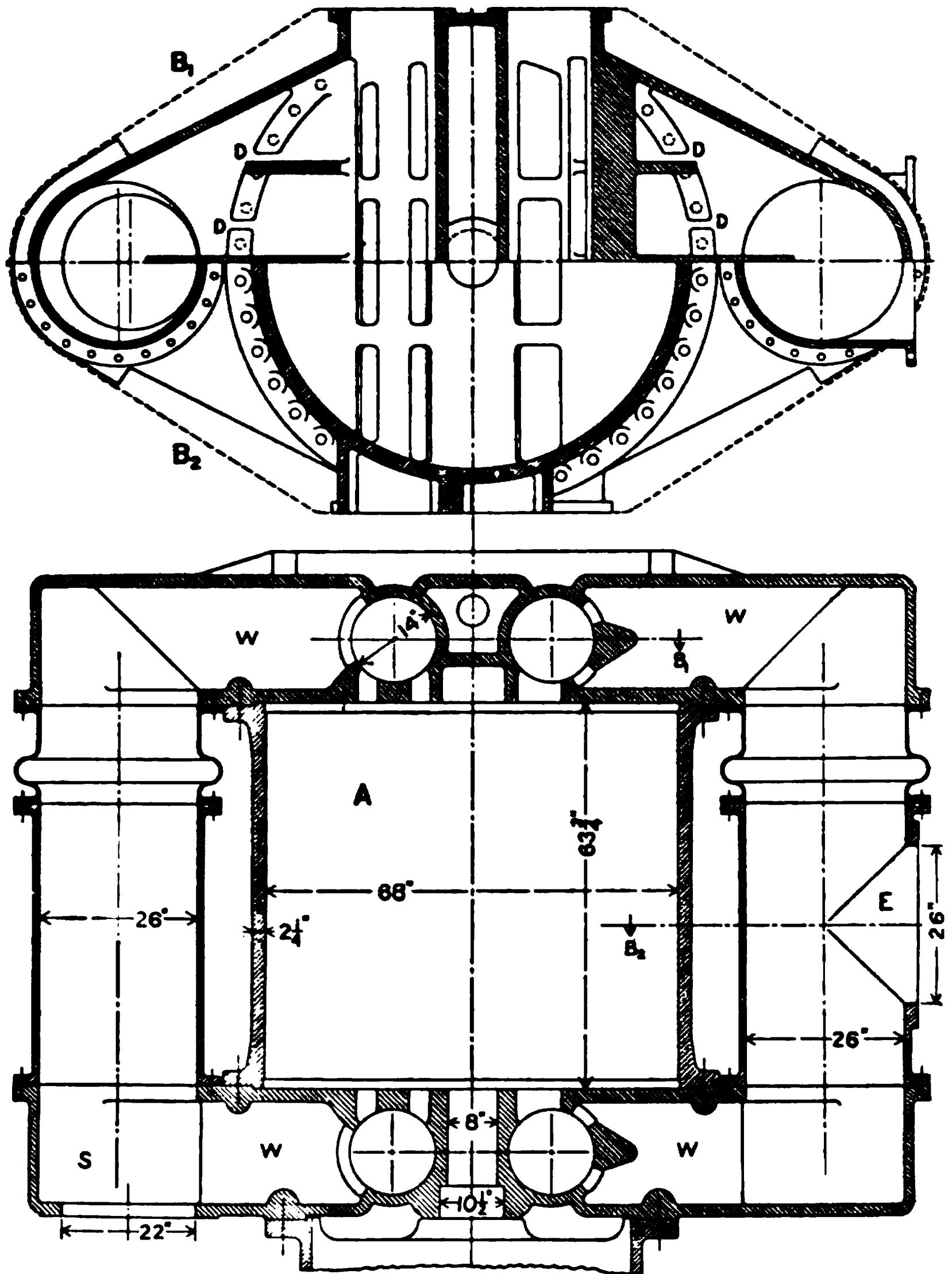


FIG. 268.—Large Corliss Cylinder with valves in heads and separate side-pipes; low-pressure cylinder of 32" and 68" by 48" vertical cross-compound engine, Allis-Chalmers Company; compare Fig. 219. Scale 1 to 36.

In Fig. 267, view A shows the most detail: we note first the form of the valve-chambers, seen also at C; next the inserted valve-seats, and the double-seated balanced valves; then the form of the steam- and exhaust-chambers, which are just like those on a common Corliss cylinder, the steam-chamber being further shown on the upper half of view B. The cylinder is very fully steam-jacketed, on both barrel and heads; a peculiar feature is the location of the throttle-valve, at the top of the jacket. It is quite usual, however, in European practice with stationary engines, thus to pass the live steam through the jacket.

This is the first engine with an extended piston-rod, running out through the back cylinder-head, that has been shown in section. External views of a similar arrangement have been given in Figs. 216 and 221; here also the outer end of the piston-rod rests upon a small slide-block. By this means the weight of the piston is largely carried by the rod, and wear between the piston and cylinder is diminished; further, the piston is a little more positively guided along the axis of the cylinder, which is the reason why the extended rod is often used in vertical, especially in marine, engines.

Some framework detail is here shown: the cylinder is supported at the head end by a broad footing cast with it; at the other end, it is carried by the frame. The position of the exhaust-valves makes it necessary that the space beneath the cylinder be kept clear, so that the foundations must come up at the sides of this space. The frame differs in detail, but not greatly in general form, from those already shown.

The valve-gear of this engine could not well be shown to the small scale used on Fig. 267: the type will be found in Chapter IX. There is a side shaft running back along the engine, as in Fig. 214, with an eccentric and a special mechanism for each valve, the steam-valves being released as in a Corliss gear. To bring quietly to rest a drop-valve, where there is a positive stoppage of metal on metal, is essentially a far more difficult and delicate task than that of the Corliss dash-pot; nevertheless, these engines are run as fast as ordinary Corliss engines, the one here drawn having a normal speed of 94 R.P.M.

(c) **CORLISS CYLINDER WITH VALVES IN HEADS.**—This arrangement is often used in large engines; for besides diminishing clearance-volume, it greatly simplifies the cylinder-casting. Fig. 268 is shown the low-pressure cylinder of a large ve

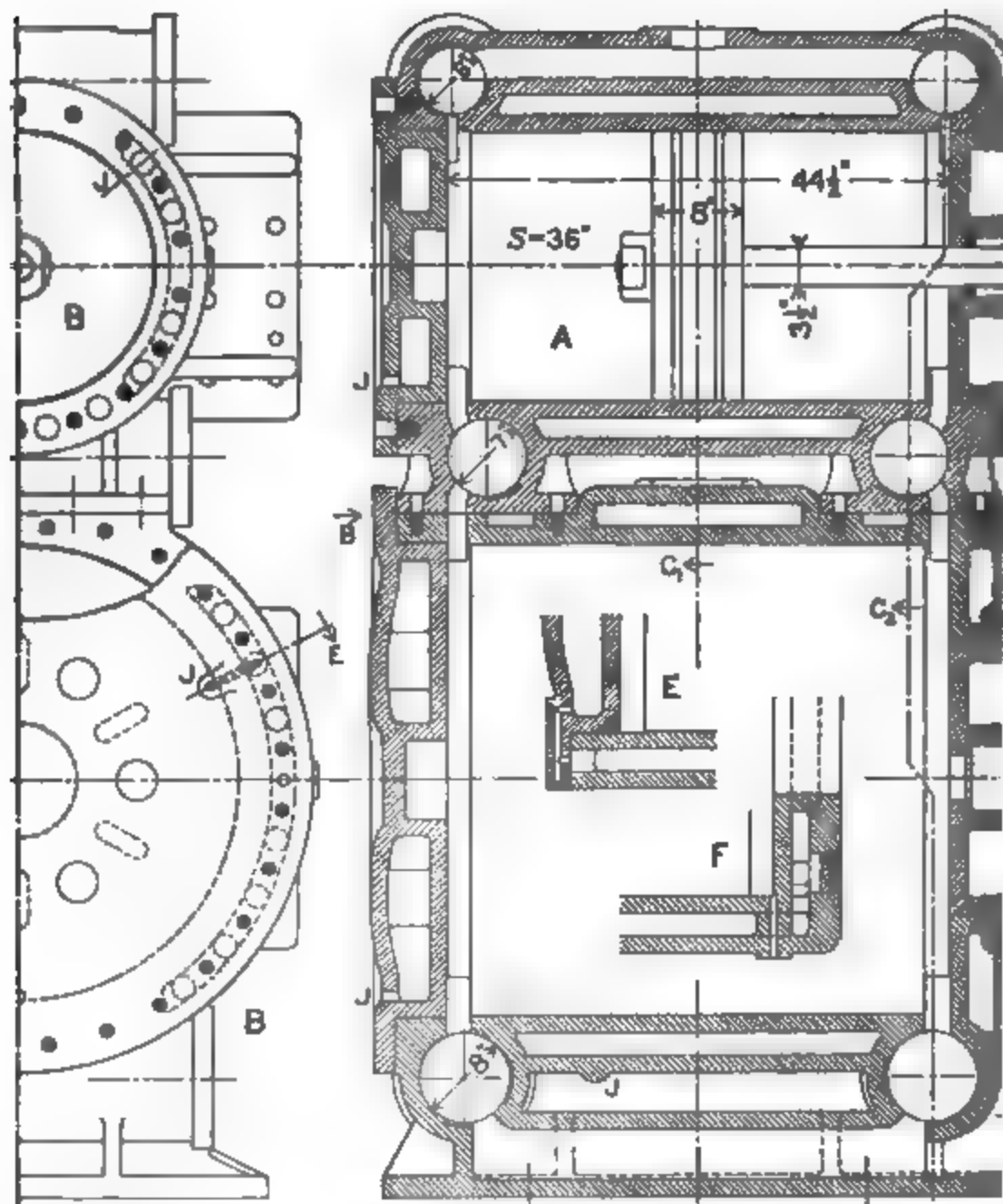


FIG. 269.—Cylinders of Holly Compound Pumping-engine, Fig. 2  
Scale 1 to 24.

engine: it is of the same design as that in Fig. 219, and the line of the covering which gives the external form is shown by dotted lines in the top view. The valves, both double-ported, are brought as close together as the piston-rod will allow, in

to get the long port-openings shown at B. The flat surfaces of the heads are tied together by several vertical webs. In the steam and exhaust connecting-pipes at the sides are included expansion sections, which are thin phosphor-bronze castings. As a

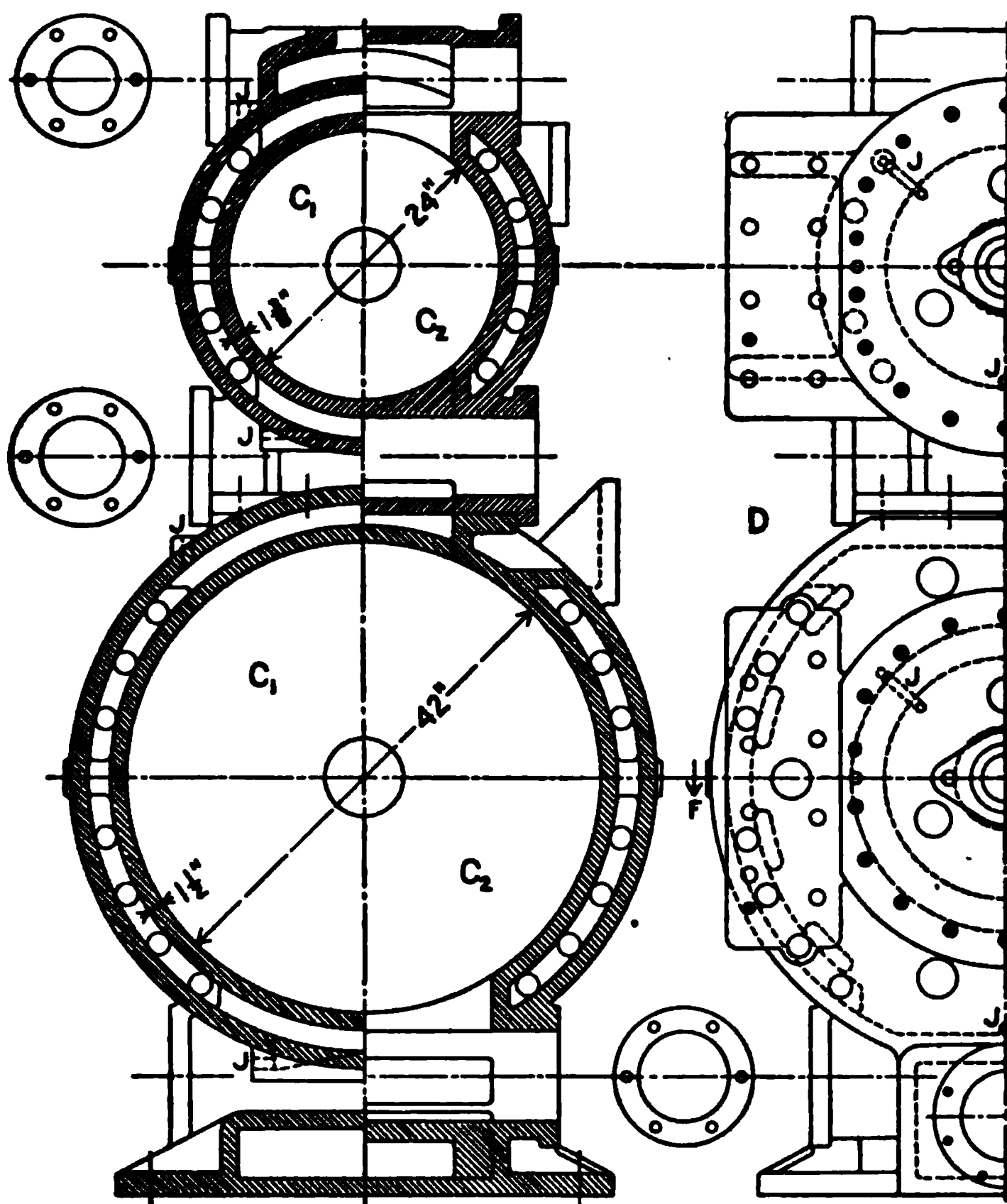


FIG. 269.—Continued.

minor detail, it is interesting to note the slots made at D, D, view B, in the ridge which gives depth of metal for the stud-bolt holes; these slots permit the drainage of the shallow pockets which would otherwise tend to catch and hold water.



Another cylinder of this type, but with the steam-chamber formed upon the cylinder-barrel in the usual manner, is shown in Fig. 271.

(p) CORLISS CYLINDER WITH SOLID CAST JACKETS.—Fig. 267 besides showing the cylinders in very compact arrangement illustrates fully the method of construction wherein all the steam-spaces are formed by coring the casting—of which Fig. 268 is a parallel example. As to the jackets on the cylinder-heads we note how they extend all around the middle of the cylinder but are kept out of the way of the valve-chambers; and how holes are formed at the ends, for digging out the sand from the casting. At the front end of the large cylinder, the side jacket runs into that on the head; the two walls being connected by struts, as shown in dotted lines on D and by the detail section F. A similar arrangement of the jacket is seen in Fig. 267.

Of the hollow cylinder-heads, only the large one on the end of the L.P. cylinder has its inner and outer walls together with webs: on all of them, the location of the core-holes is shown. They all receive their steam-supply from the jackets after the manner shown at detail view E. On this as also on Fig. 267, the letter J is used to mark the points of connection, whether for supply or drainage. While not necessary it is better to plug all the core-holes in the ends of the cylinders except those concerned in the jacket-supply, in order to diminish the number of places where leakage is likely to occur.

(q) CYLINDERS WITH LINERS.—The decidedly more compact method of construction for steam-jacketed cylinders, especially at large sizes, is exemplified in Figs. 270 to 273. When the cylinder-wall proper is thus a separate piece, the operation of making the casting, in the foundry, is very considerably simplified; it is easier to get a sound casting with the simpler form; and the metal for the liner can be especially adapted to resisting wear. In Fig. 270, where two half-sections at right angles to each other are swung into the same plane, the exhaust-valves are shown; but a special feature of the design is the manner in which the exhaust-chamber surrounds the cylinder, outside the jacket, so that the latter serves also, to some extent, as a reheater.



exhaust from this H.P. cylinder passes from the outlet at the left, through a connecting-piece, to a similar inlet on the L.P. cylinder. The cylinder-heads are jacketed, and in each there are two relief-valves.

When a liner is used in an engine-cylinder, provision must be made for carrying the steam-ports through or past it, and for holding the liner in place. In Fig. 270 the ports pass the ends

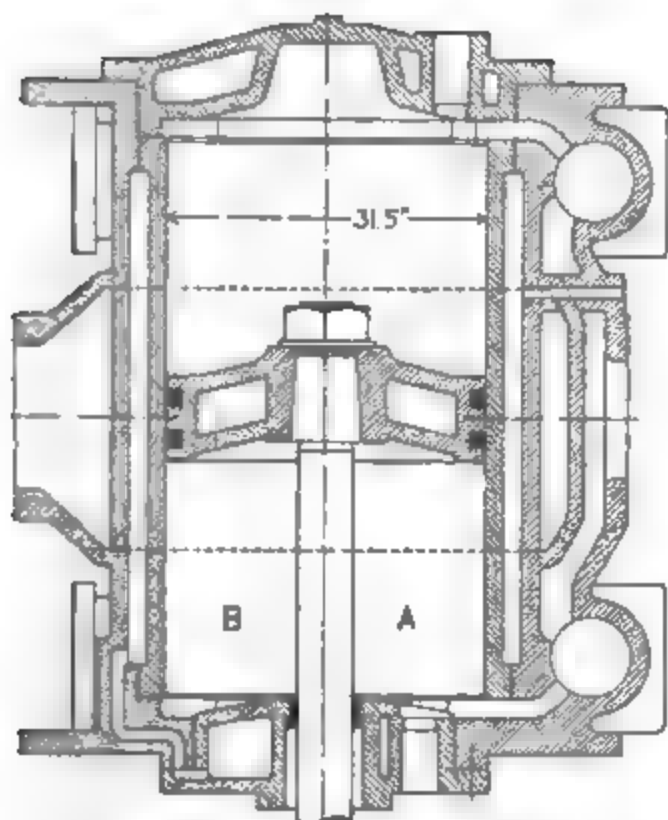


FIG. 270.—Two Half-sections of High-pressure cylinder of French vertical Corliss engine, see *Engineering*, 1901, Feb. 8. Scale 1 to 80.

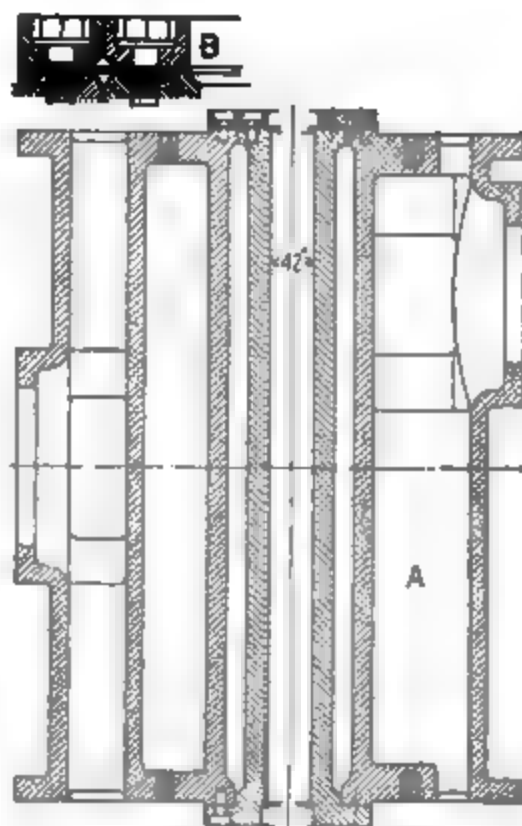


FIG. 271.—Corliss Cylinder, general arrangement as in Fig. 268, but with liner and with steam-chambers cast on cylinder. Scale 1 to 80.

of the liner; and between the ports there are projecting shoulders upon the cylinder-heads, which touch the ends of the liner and keep it in position. Fig. 271 gives two partial sections of a cylinder arranged as in Fig. 268: here the ports are entirely clear of the jacket; the liner is bolted fast at one end, and at the other end is used a diaphragm joint, made with an annulus of thin copper plate held down by two rings—a method used in Fig. 266 also. Of course, the cylinder-heads are formed around these jacket-joints, without clearance.

In Fig. 272 we have a case where the ports pass through the liner, which must then have a long bearing at each end; and we see further an example from American practice wherein the steam passes through the jacket on its way to the engine. It does appear, however, that this jacket is to be considered rather an adjunct to the large steam-passage which rises from the inlet at the bottom than itself the steam-passage as in Fig. 267; and that most of the water formed by condensation in the jacket will have a chance to separate quietly and be drained away, without being caught up by the main current and carried into the cylinder.

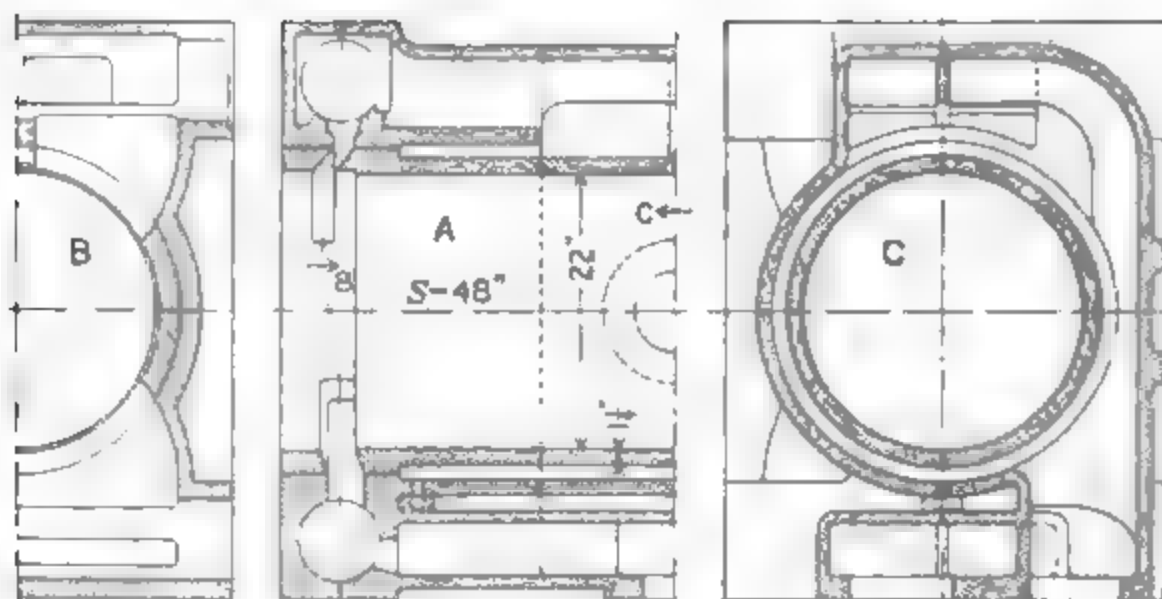


FIG. 272. —High-pressure Cylinder of Corliss Air-compressor. Scale 1 to 24.

The most usual form of liner for a marine cylinder is shown in Fig. 273 I.: the flange at the bottom is tightly fitted and fastened with sunk-head bolts: at the top the plain contact joint is reinforced by a soft copper ring, which is calked into the dovetail groove. In II., with the same general arrangement, a packing-ring is used, held down by screws upon a filling of hard-packed asbestos in the recess. At III. is seen a special recessed joint at the bottom; while IV. shows the occasionally used method of bolting the liner fast at the top, with bolts spaced as far around the cylinder as the steam-port will permit.

The liner is forced into place, usually with a hydraulic press; but to make the fit close enough to prevent leakage is likely to

bring too much strain upon the outer shell, especially with the thin castings usual in marine construction—hence the employment of special devices for making the joints tight. For a fixed joint, as at the bottom in Fig. 273 I., a coating of red-lead paste

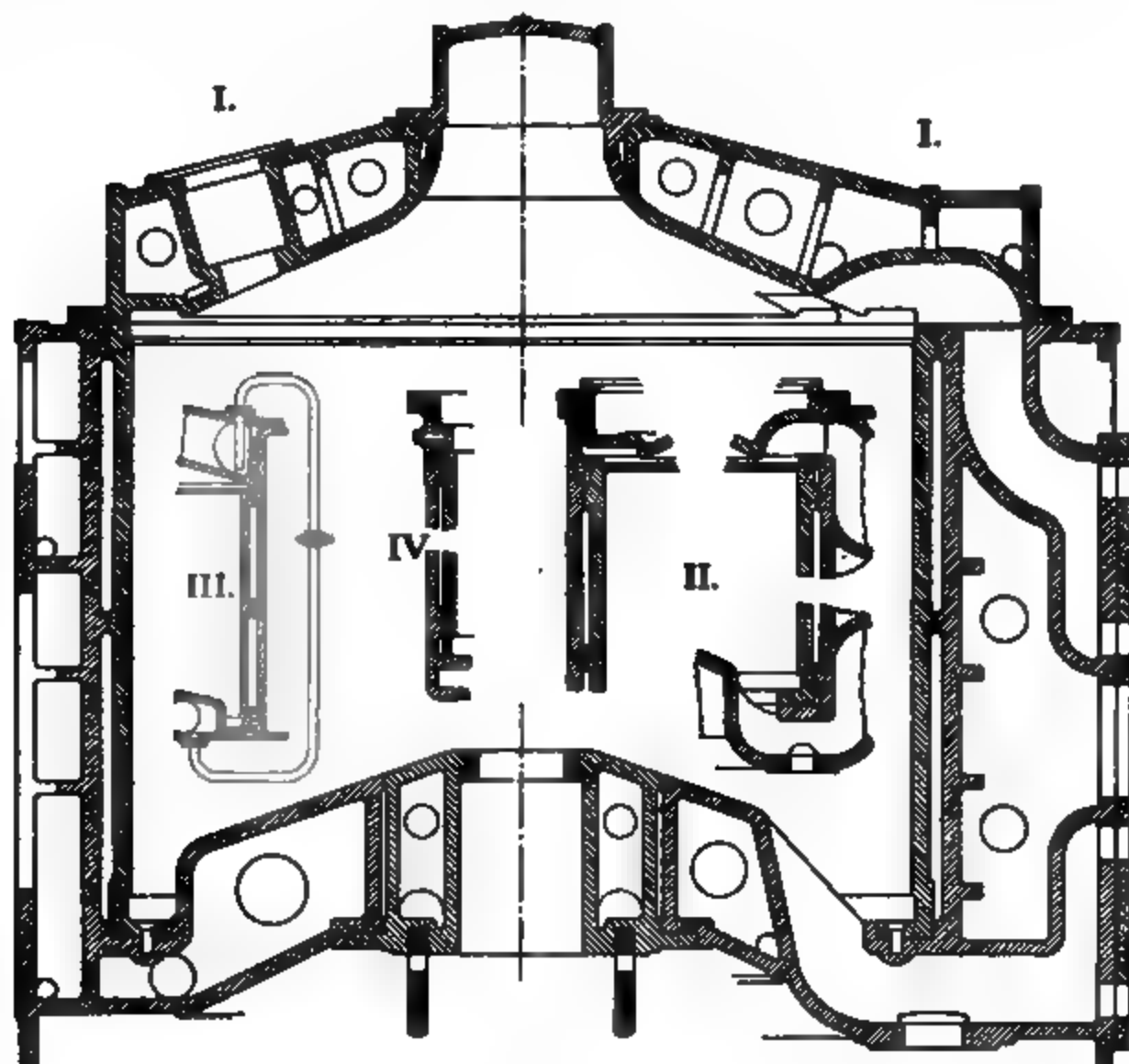


FIG. 273.—Marine Cylinders with Liners.

I. French Battleship, III. French Torpedo-boat, both by Schneider & Co., see *Engineering*, 1898 II; II. and IV. U. S. Cruisers. Scale 1 to 24.

between the surfaces is very effective; but where variable expansion of the two shells causes relative movement this can do little good. The successive cylindrical fits are, of course, made larger in diameter toward the top, so that the liner can be dropped almost into place, and need be forced home through only a short distance. The contact band at mid-length is usually continuous, except

where grooves are cut through the ribs after machining, to the steam pass.

Of other important features shown in Fig. 273, the first is cellular form of the heads in I. and III., which are made of double walls fully braced, as much for strength as for the sake of the steam-jacketing. In I., further, we see the cylinder cut off short at the top, and the steam-port formed partly in the head; this simplifies the main casting but gives a non-circular bolting-flange. At III. is shown the method of draining the head when used as a steam-jacket: since the drain-pipe serves as supply to the lower head, condensation in the latter makes effective the siphon, which draws out any water that may collect at the lowest part of the upper head.

Sometimes stationary or locomotive cylinders are fitted with liners in the form of a bushing with a continuous fit, an example appearing in Fig. 438. This permits the use of a hard, close-grained iron for the rubbing-surface. The liner may be a part of the original design, may be added where the main casting has shown blow-holes or sponginess, or may be used to reduce the diameter after extensive re-boring. Always in stationary practice, and usually in marine, the jacket-liner is of cast iron; but sometimes forged steel is used, especially in naval engines where the saving weight is highly important.

(7) STRENGTH OF THE CYLINDER.—(Of the various forces which act upon the cylinder, only the internal steam-pressure produces stresses which can be calculated; but the other, unknown force-actions may have a large effect, so that the designing of the cylinder is very much a matter of experience and good judgment. To get an idea of the effect of steam-pressure alone, apply to several of the examples illustrated the formula for stress tending to cause longitudinal rupture, namely,

$$PD = 2St; \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad . \quad (3)$$

where  $P$  = steam-pressure, in pounds per square inch;

$D$  = diameter of cylinder, in inches;

$t$  = thickness of metal, in inches;

$S$  = tensile stress, pounds per square inch.

Results got by this formula are given in Table 43 A, where the stated or assumed highest probable boiler-pressure is used. The first cylinder is exceptionally thick, other designers using about 1" for this size. It appears, then, that in stationary engines *S* is likely to be somewhere in the neighborhood of 1000 lbs., while for transportation service it ranges from 1500 to 1700. In the last two examples, a calculation thus based on the least thickness is not at all a true test of strength, the casting being strongly reinforced by ribs and flanges.

TABLE 43 A. STRESS IN CYLINDER-WALLS  
DUE TO STEAM-PRESSURE.

Fig.	Type of Engine.	Pressure.	Diameter.	Thickness.	Stress.
251	High-speed. . . . .	100	15"	1½"	600
262	Corliss, simple. . . . .	100	26"	1½"	1040
268 (H)*	" compound. . . . .	150	32"	2"	1200
271 (H)	" " . . . . .	150	42"	2"	1575
247	Locomotive. . . . .	200	20"	1½"	1600
245 (H)	Marine. . . . .	180	17"	1"	1530
259 (H)	Naval. . . . .	200	14.5"	¾"	1660
273 I. (H)	" . . . . .	210	44.5"	1.57"	2970
273 III. (H)	" . . . . .	210	16.7"	.53"	3300

SEATON gives for the minimum thickness of marine-engine cylinders

$$t = \frac{PD}{3000} \text{ to } \frac{PD}{3600}, \quad . . . . . \quad (302)$$

or *S* from 1500 to 1800, the latter figure being for the highest grade of iron.

While the body of the cylinder is of a self-sustaining form, other parts, as for example the walls of the steam-passages, need support. In marine cylinders especially, free use is made of cross-webs: and in some cases even stay-bolts or stay-rods, analogous to those used in boilers, are put in to tie the port-walls together at critical points—see Figs. 411 and 414.

\* The symbol (H) means that the high-pressure cylinder of the engine, whether shown in the drawing or not, is the one referred to.

Other stresses besides those due to pressure may be caused by irregular contraction in the original cooling of the casting, by unequal temperatures of the different parts under working conditions, by external forces due to the manner of support or connection to the framework, and especially by the accidental presence of water. The most frequent cause of breakage of the cylinder is the effort of the piston to compress into the clearance space a body of water larger than the volume of that space. Preventives are, a well-drained separator in the steam-pipe, liberating drains and relief-valves on the cylinder, and, with a jet-condenser, a vacuum-breaker which will admit air if the condenser accidentally fills with water up to a certain level and thereby keep the water from ever being sucked up into the cylinder.

An important consideration is that a fairly thick cylinder can be molded with less trouble and cast with a better expectation of soundness of metal than one that is relatively much thinner. Take Fig. 273 I. as an example: it is very fully ribbed and braced, which makes the pattern expensive; there will be many cores in the mold, and they must be very accurately set and firmly held, and a high degree of skill is needed to get a sound casting. For all these reasons, the saving of weight in marine cylinders is far from representing a saving in cost. This requirement of a good thickness of metal, not only for constructive reasons, but also to permit re-boring of the cylinder when worn, is embodied in empirical formulas like that of UNWIN,

$$t = .02D + 0.5 \text{ to } .02D + 0.75 \text{ ins.}; \quad . \quad . \quad . \quad (303)$$

where the effect of the constant term is to make a small cylinder relatively thicker than a large one.

In the matter of external forces, the cylinder that is supported wholly or chiefly by being bolted to a full circular seating on the frame is least likely to be unduly strained. That is, in the usual Corliss type of construction, where the relatively large weight of the cylinder is carried by the foundation, there is a possibility that irregular settling of the latter may cause severe stress; while the overhanging cylinder of the small high-speed engine, once

made strong enough to carry its own weight, is not liable to an increase in this load. In vertical engines, the complete frame of the stationary type carries the cylinder in a more favorable manner than does the marine frame. Large low-pressure cylinders, relatively thin, are not at all rigid, so that even their own weight can distort them; thus the larger cylinders for a marine engine must be bored with the axis vertical if they are to be truly round when mounted in place upon the engine.

It is not the purpose of this book to go into the details of the design of the machine for strength: except as an idea of proportion may be got from the numerous drawings given, the reader is referred to works on machine design for rules covering these matters.

(s) STEAM-TIGHT JOINTS.—An important detail of the flanged cylinder-head joint, shown with most emphasis on Figs. 247 and 256, is the formation of a narrow contact-ring inside of the bolts. The rest of the flange-surface is cut back a little, or relieved, so that the full grip of the bolts is concentrated on this narrow strip. This surface is finished true and flat, and constitutes a “ground” joint, with which no packing need be used. When packing is found necessary, a gasket of heavy paper is often enough, with good surfaces; sometimes metallic gaskets, made of lead or soft copper, are used; but rubber sheet-packing, such as is placed between pipe-flanges, ought not to be required to make any cylinder-joint tight.

Any arrangement other than the firm contact of metal on metal is especially out of place when the flange of the front cylinder-head comes between the cylinder and the frame, as in Figs. 241, 250, 251, etc. The double requirement of making a tight joint and of holding the cylinder in true alignment is somewhat more severe upon the front bolts than is that of merely holding the cylinder-head—this depending very much, however, upon how truly and strongly the cylinder is supported—and quite often these bolts are made a little heavier than those at the back. In Figs. 254, 260, 263, 267, etc., are seen the examples in which the front head is cast solid with the cylinder; nearly always, with this arrangement, there is an enlarged opening through which a

heavy boring-bar can pass, and which is then closed by the inserted stuffing-box. In Fig. 253 the front head is put into place through the cylinder, and is held fast by a few light stud-bolts, which are helped by the steam-pressure. The narrow joint surface, at both ends, is packed by a ring of rectangular copper wire, forced into a groove in the face of the ledge on the head, and faced off so as to project about  $\frac{1}{4}$  in.

*Bolts.*—It will be noted that stud-bolts seem to be used altogether; rarely, if ever, are the flanges made wide enough to give room for through-bolts, nor is it desirable to give the bolt-circle the needlessly large diameter that this would require. The stress in these bolts is a resultant of the original tension due to the screwing up of the nuts and of that due to steam-pressure. When the bolts are screwed tight, they are stretched (within the elastic limit, of course) and the metal of the contact surfaces is compressed; when the steam-pressure comes on the head, it stretches the bolts still farther, and in so doing eases up somewhat the pressure in the joint. The total tension in the bolts is, therefore, not the original tension plus the steam-pressure, but is less than this sum. It is usual, however, to base the effective sectional area of the bolts upon the total steam-pressure on the head, using a low working stress in order to allow for the rather uncertain initial tension. This working stress is usually from 3000 to 4000 lbs. per sq. in. and the distance between the bolts, which determines their number, varies from 4" to 6", depending upon the thickness and stiffness of the flange.

The considerations just stated apply also to the joints under the various valve-chest covers. These pieces must, of course, be designed so as to have the requisite strength and stiffness to resist the pressure to which they are subjected.

A minor detail of considerable usefulness is the jack-, or lifting-screws, for starting cylinder-heads and other bolted covers when taking them off. At the ends of a diameter two holes are tapped in the flange, between the bolt-holes; and tap-bolts run through these against the cylinder-flange loosen the head very easily, after which the same holes can be used for the lifting-handles or eye-bolts.



(t) CYLINDER PROPORTIONS.—The ratio of diameter to stroke varies with the different classes of engines. In the short-stroke, high-speed type it is usually a little less than one, but is occasionally greater; in simple Corliss engines the ratio is generally from 0.5 to 0.6. Multiple-expansion engines begin with some such ratio as this, but often get it well up toward 2.0 in the low-pressure cylinder. The ratio of diameter to length (between heads) would give a better idea of the real form of the cylinder; but this would involve the thickness of the piston, which varies quite widely.

The piston-clearance, or the distance between cylinder-head and piston when the latter is at the end of the stroke, depends upon the size of the engine and upon the number of working joints, subject to wear and adjustment, between the cylinder and the shaft-bearing. Thus one rule is, for small engines start with  $\frac{1}{8}$  in. and add  $\frac{1}{16}$  for each joint; for large engines, change these values to  $\frac{1}{4}$  and  $\frac{1}{8}$ . If the pistons and heads are faced off and if the joints are so arranged that their take-ups neutralize each other (see Figs. 302, 304), much smaller clearance-distances may be used.

(u) DESIGN OF THE STEAM-PASSAGES.—This is a simple matter that can very well be taken up here, without waiting for the close study of the form and action of the valve. The determining factor is the permissible velocity of the steam-current. For a given rate of flow, the area of cross-section must be such that this velocity will not be too high. Rate of flow and rate of piston-displacement have the same factors, area and velocity; and if the steam is to fill up closely the space behind the moving piston, these two volume-rates must be equal.

It is customary to proportion the steam-passages so that the maximum velocity of flow shall not be greater than from 160 to 240 ft. per sec.; this maximum occurring when the piston is near the middle of its stroke, and therefore seldom existing, for the entering steam, in engines where the normal cut-off is early. During exhaust, however, this maximum velocity is always reached. It may appear that the exhaust steam will have to flow much more rapidly than the live steam because its specific volume is so much greater; but we must remember that a great part of the steam escapes while the piston is moving slowly, near the end of the

stroke; and thereafter the steam of low pressure has merely to get out of the way as the piston returns.

Referring to § 26 (b), we see that the pressure-drop required to produce the above-named velocities, through an ideal orifice would be very small—far less than the loss between the steam-chest and cylinder usually observed in engines. But other resistances have a greater effect than does the inertia of the steam, and this range of steam-velocity has been found by experience to embody a good compromise between excessive pressure-drop on one hand and excessive clearance-volume on the other.

Comparing the maximum velocity of the piston (essentially the same as the linear velocity of the crank-pin) with the mean rate of travel, we find the ratio to be 1.6 to 1; with infinite connecting-rod the ratio is  $\pi$  to 2 or 1.57 to 1; and the effect of the actual rod is closely enough accounted for by using 1.6. With this ratio a maximum velocity of 160 to 240 corresponds to a mean of 100 to 150 ft. per sec., or of 6000 to 9000 ft. per min. The last is directly comparable with the mean piston speed of 600 to 1200 ft. per min. and the section of the steam-passage ranges, in practice, from one-tenth to one-sixth of the area of the piston, with one-eighth as a good average value.

After the velocity, the next consideration is the desirability of keeping down the amount of clearance-space: this exerts a strong influence in the direction of making the ports short and direct by the use of separate valves near the ends of the cylinder. When there are separate valves for admission and exhaust, and the prevailing cut-off is early, it is customary to make the exhaust port larger than the other, as appears in Figs. 262, 264, 269, etc.

#### § 44. The Reciprocating Parts of the Engine.

(a) THE PISTON.—Before reviewing the pistons which have appeared in the cylinder-drawings just given, and before discussing the more detailed drawings now to be presented, it may be well to re-state the conditions which this piece must meet, as already set forth in § 2 (c):

*First*, the piston must have strength to resist the steam-pressure; but it must be no heavier than necessary, lest there be excessive inertia of the moving parts and undue wear of the cylinder.

*Second*, the piston must have a broad rim or working face, especially in horizontal engines, to furnish plenty of rubbing surface and diminish wear.

*Third*, tightness against the leakage of steam must be secured, by the use of packing-rings. The solid piston-body is a little smaller than the cylinder-bore, and the gap between them is closed by the packing.

(b) SOLID BOX PISTONS.—Sectional views of the plain hollow cast-iron piston, all in one piece, have been given in Figs. 6, 247, 251, 254, 256, and 267, so that no further illustration of this type is needed. For diameters less than 24 inches it is overwhelmingly prevalent; and it is often used for larger diameters, up to 48 inches. An absolutely simple example is given in Fig. 254, where the hub is cylindrical, the two disk-faces are flat and of uniform thickness, a number of radial stiffening-ribs are used, and there are two plain packing-rings. In Fig. 247 the design is closely similar except that the faces are made very slightly conical, so as to taper the metal toward the rim, and the nut is partly sunk into the hub. In Fig. 251 the piston is unusually broad, the rim is reduced in thickness between the slots, and no radial ribs are used. The noteworthy feature in Fig. 267 is the close accommodation of the outer profile of the hub-section to the shape of the hole for the rod. Fig. 6 shows a very light piston, with a special method of securing the rod and a large number of very light packing-rings: this piston is made with the metal thin not merely to save weight, but also with the idea that the piston, easily replaced, will be the part most likely to break in case of a smash-up due to the accidental presence of water in the cylinder. Finally, in Fig. 270 we have a box piston made slightly conical, which is a step in the direction of marine practice.

Comparing Figs. 247 and 254 with 6 and 267, we distinguish two types of ribs: the first comes out to the rim, with a hole near the middle of the rib; the other stops short of the rim. All these

pistons come from the foundry with a number of core-holes in the face, usually one hole to each of the pockets formed by the rim. These holes are tapped and plugged before the piston is finished.

(c) A GROUP OF LOCOMOTIVE PISTONS is given in Fig. 274. This is a favorable field from which to select examples, because there is a wide variation in type of construction with model size. Types I., II., and III. are from European practice, where the box piston is less usual than in American locomotives. The flat central-disk design, No. I., is an often used English type; the side-disk piston at II. is drawn accurately to scale from a standard German locomotive; while III. shows the cone-disk adapted from the marine engine. A true marine piston would have the disk come to the other side of the rim, as will be seen

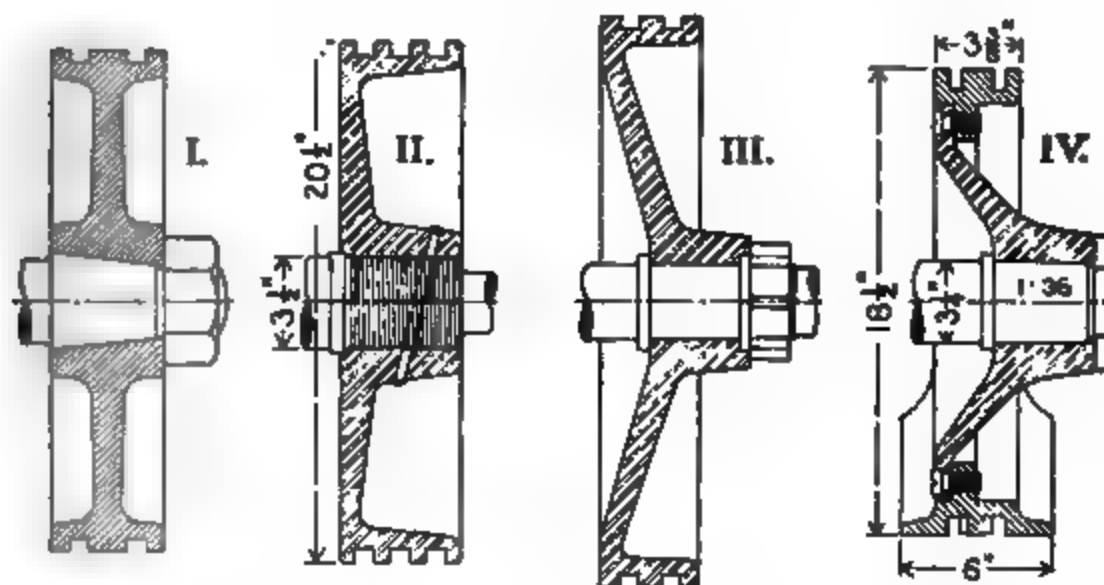


FIG. 274.—Locomotive Pistons of the Solid-disk Type. Scale 1 to 12

turning this drawing into the vertical-engine position, and then comparing it with Fig. 260 or Fig. 279. A still further departure from the marine type is seen in Fig. 256, where the larger piston has a Z-shaped section.

As drawn, Fig. 274 I. is heavy enough to be made of cast iron, but II. and III., as also the L.P. piston in Fig. 256, are cast-steel patterns. American designers have been chary of bringing cast steel into rubbing contact with cast iron, believing that there is a great liability to excessive wear and "cutting" of the surface. In Fig. 256, for instance, a brass facing is cast upon the rim of

piston, covering the whole space between the packing-rings (see also Fig. 280 III.): and in a similar way, a broad, thin cast-iron ring, made in halves, has been riveted upon the rim of a piston of this form. In Fig. 274 IV. is seen an arrangement which combines the steel cone-disk for strength with a cast-iron rim for wear. This is a standard design on the Pennsylvania Railroad, and is made in several diameters with a rim of the same width. The latter is broadened at the bottom over 120 degrees of its circumference, so as to have more weight-carrying surface. The rim is made .01 in. smaller than the disk at the fit and shrunk on, and the screws are riveted as shown. An advantage of this separate construction is, that when a cylinder is re-bored, only the cast-iron ring, and not the more expensive steel casting, need be renewed. It is to be noted that both the all-steel designs, at III. and II., have the extended piston-rod, which helps to carry the piston, and also steadies and guides it so as to diminish wear.

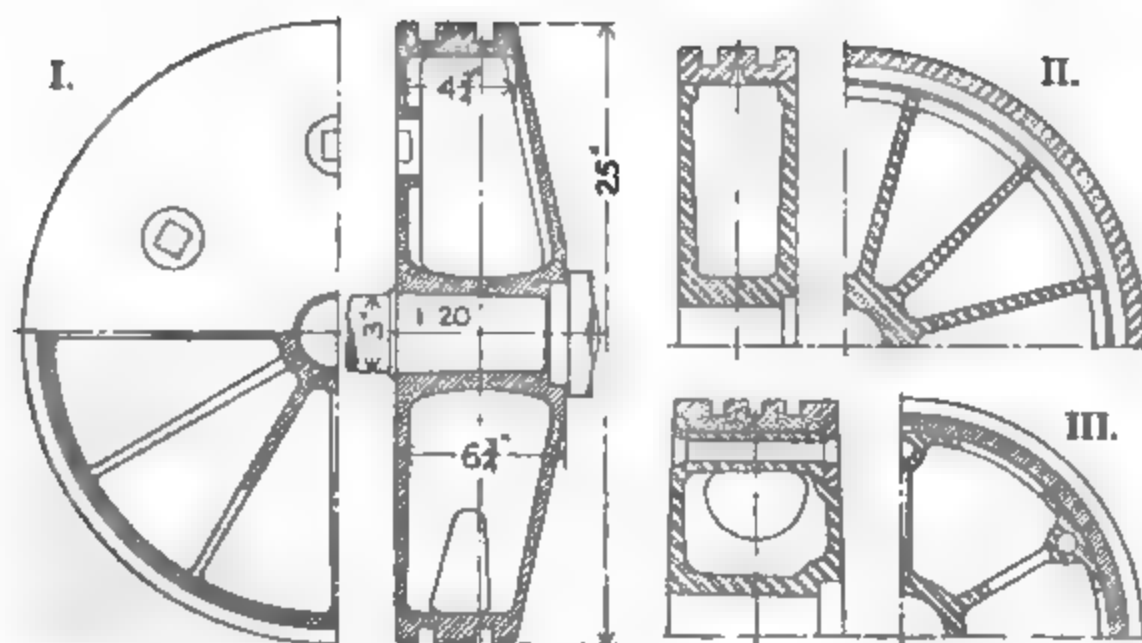


FIG. 275.—Extra Light Hollow Pistons for Locomotives. Scale 1 to 12.

The drawings in Fig. 275 show special construction of the box piston. I. is a standard Baldwin Locomotive Works design for the low-pressure cylinder of compound engines: as here shown, extra light, it is made of malleable iron (decarbonized casting); for plain cast iron, the cores are made smaller, so as to increase the thickness of metal by about 60 per cent. In the rim of this

piston are several small grooves filled with babbitt-metal, make a better rubbing surface.

Drawings II. and III. show cast-steel box-pistons. In general a hollow steel casting must have larger openings for "venting" and for removing the cores than are necessary with iron. In then, the body and rim are made separate, and put together by a screwed and riveted joint. In III. the rim is a cast-iron rim held in place by a narrow steel follower-ring, which may be riveted as here, or bolted fast. This design resembles Fig. 274 IV. in case of renewing the rim.

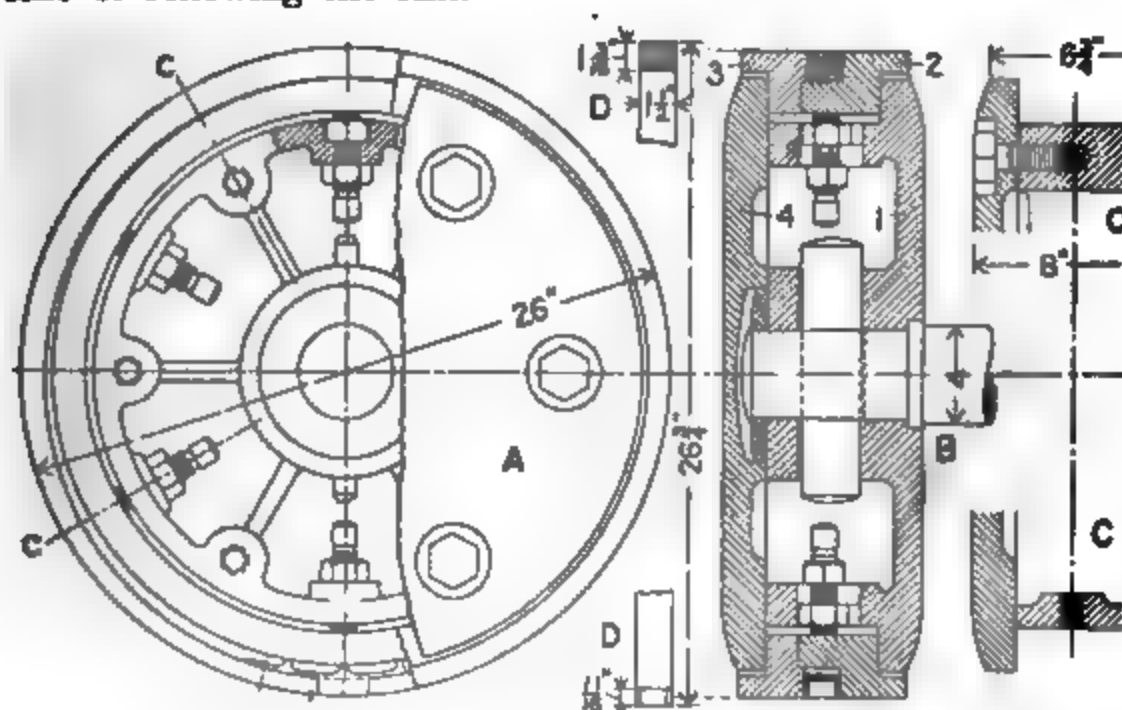


FIG. 276.—Built-up Piston, for Corliss Engine in Fig. 210. Scale 1 to

(d) BUILT-UP BOX PISTONS.—A type of piston which long been used in engines of the Corliss class is well represented by Fig. 276. In this there is first the ribbed body or "spider" into which the rod is securely fastened; then the rim or "bull-ring" or "junk-ring," which carries the packing-ring; last, the follower plate 4, held fast by six tap-bolts. The bull-ring forms the working surface of the piston, and can be adjusted by set-screws so as to make the axis of the piston agree with that of the cylinder. The bull-ring is made in two parts, 2 and 3, partly because the packing-ring is rather too stiff to be "sprung" over the piston in order to get it into its grooves, partly because access can now be had to the pack

ring without withdrawing the piston from the cylinder-bore. The rod has a taper fit against a collar, is held by a cotter, and is riveted at the end, over a heavy washer: note how the slot for the key cuts through the rim of the piston-body at both sides, making necessary a special arrangement of the set-screws on this diameter. The packing-ring is of the tapered form, and the "keeper" which joins its ends is more fully shown in Fig. 280 IX.

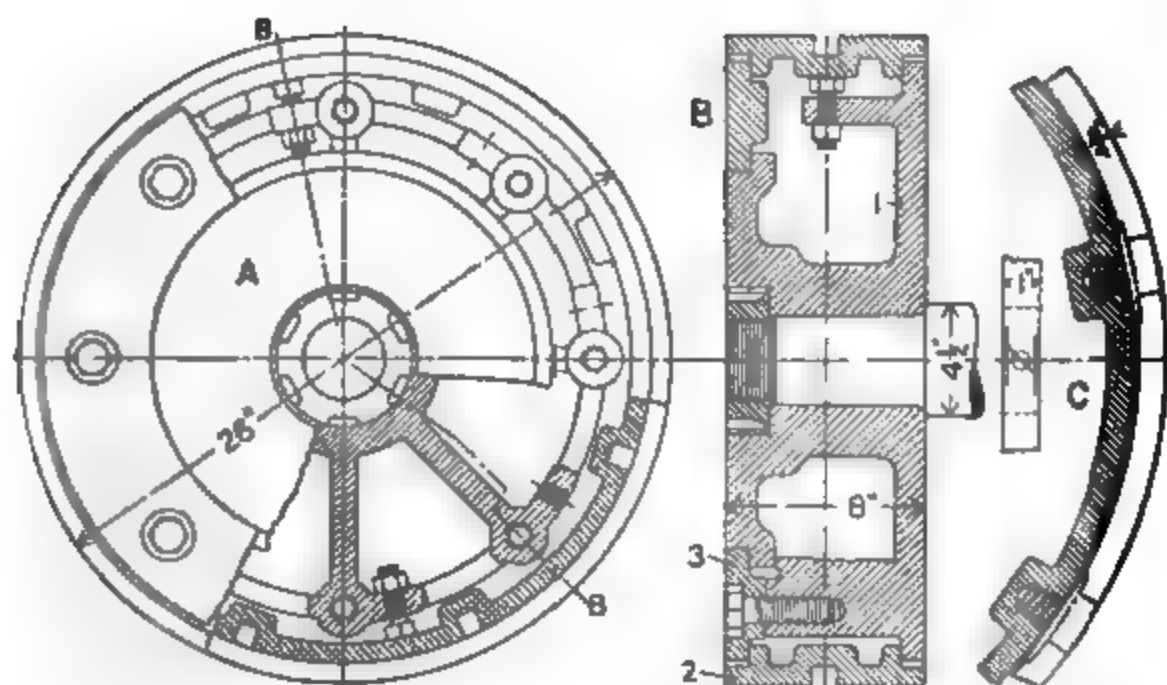


FIG. 277.—Piston for 26" by 48" Corliss Engine, Figs. 218, 241.  
Scale 1 to 12.

In Figs. 277 and 278 are given further examples of the type just described, which sufficiently represent the possible variations in form. Fig. 277 has the bull-ring all in one piece, with set-screw adjustment, and only an annular follower-plate. The packing-ring is made in segments and pushed out by a spring under each of the keepers which form the joints. In Fig. 278 the follower forms half of the bull-ring, the two halves fitting together tightly with a recessed joint, but having considerable play on the piston for adjustment. The latter is determined by the two little blocks marked 4 on view A: of the seven slots in the ring 2, only two are thus filled, at the bottom where the blocks will carry the weight of the piston; and the screwing-up of the follower-bolts grips the whole structure tightly together. Here a continuous packing-

ring (with one joint) is used, and its elasticity is supplemented by flat springs as shown at C.

The purpose of this latter design is partly to make the packing rings removable without disconnecting the rod from the crank head or disturbing the alignment of the piston in the cylinder. For this purpose, the follower, as also the packing-ring and the little blocks 4, 4, have tapped holes in them for the insertion of eye-bolt handles. Always the piston is provided with this means of getting a good hold when taking it out of the cylinder—as shown for instance, in Fig. 6.

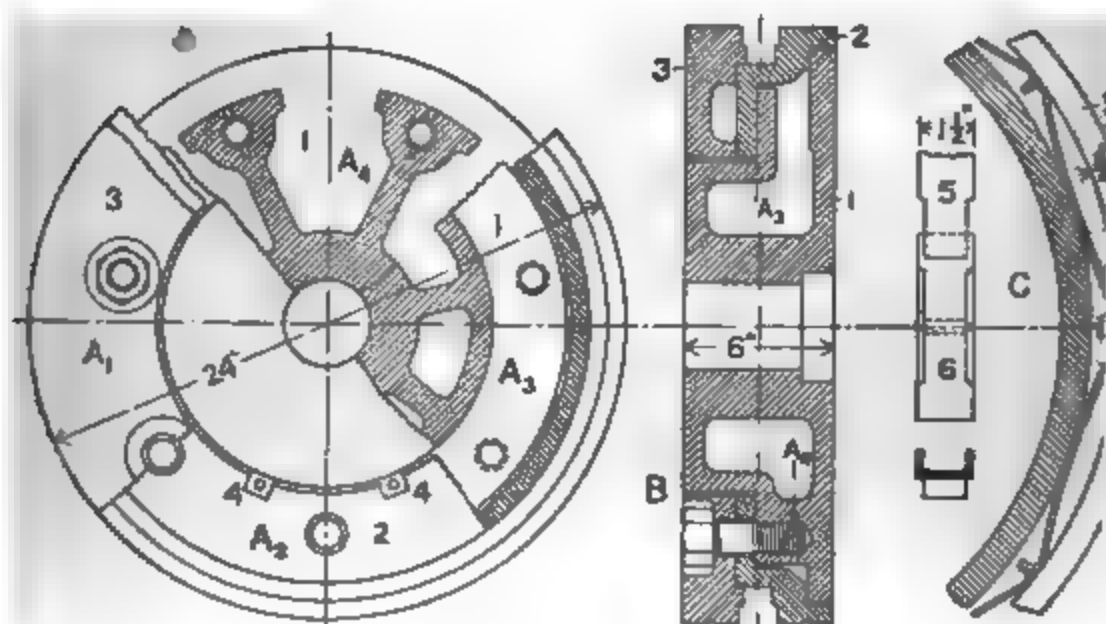


FIG. 278.—Reynolds-Corliss Piston, Allis-Chalmers Company.  
Scale 1 to 12.

It is important, as a matter of practical convenience, that the follower-plate be easily removable, or that the bolts shall rust fast. Steel tap-bolts in cast iron, or steel nuts on studs, will be satisfactory if the joint is opened at short enough intervals. Frequently, however, a screw-joint with bronze on steel is secured either by the use of bronze nuts on stud-bolts, or by screwing tap-bolts into blocks of bronze which are let in from the side recesses in the piston-body. Note how the studs are secured by pins in Fig. 278, so that they will not come out with the nuts.

(c) SOLID-DISK PISTONS.—Simple cone-disk pistons of the marine type are shown in Fig. 260. Usually the single-disk piston is used in a vertical engine, and especially with an exte



piston-rod, has a much narrower rim than a piston of the same diameter would have in a horizontal engine. An extreme case is seen in Fig. 279 I., where the packing-rings, self-elastic, form the

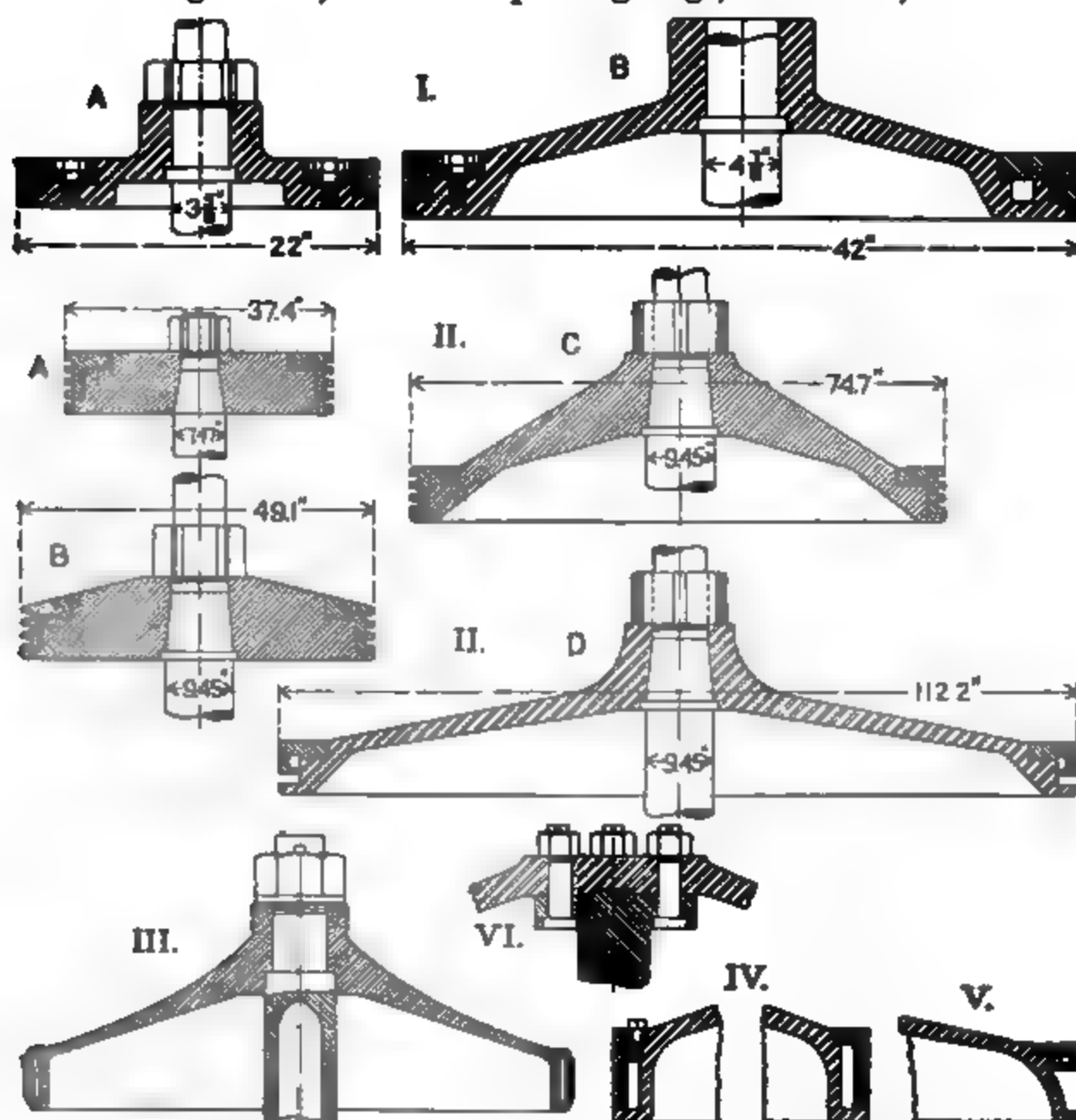


FIG. 279.—Pistons of the Marine Type.

I. Vertical compound Buckeye engine, 27" stroke, scale 1 to 16; II. Pistons from one of four engines on the steamer "Kaiser Wilhelm II.," with 70.8" stroke, scale 1 to 86; III. Forged steel piston, about 24" in diameter, for torpedo-boat; IV., V. Saving weight in the rim; VI. Special form of rod connection.

whole surface of the rim. These pistons are steel castings, with follower and spring-rings of cast iron.

The group of pistons in Fig. 279 II. belongs to a large three-

crank quadruple-expansion engine, with the high-pressure cylinder placed tandem above the first intermediate. The smaller pistons are made thick and heavy as a part of the scheme of balancing the engine. Note the arrangement of the combined bull-rings and followers, whereby all the packing-rings can be removed without disturbing the pistons. On the first three pistons, plain snap-rings are used; the low-pressure has a flexible (segmental) ring, backed by springs, of the type shown in Fig. 280 XI.

In Fig. 265 is seen a piston which may be considered a development from the marine type, being made of two cone-disks combined into the hollow, box form and held together by the nut on the piston-rod. This method of making a hollow piston in two pieces is much better, with cast steel, than to try to core it out. Another example of the same general type of construction is partly shown in Fig. 281 III.

The most usual type of piston in large stationary engine is probably the box form, with parallel faces and made of cast iron, the rim arrangement being of the type in Fig. 278, modified to the simple device used on Fig. 279 II.

The extreme of lightness is seen in the forged-steel piston at Fig. 279 III.; and rim-sections without any dead weight, for marine pistons, are shown at IV. and V., both from naval engines. Figure VI., is illustrated a peculiar method of fastening piston and piston-rod together, which has been used in small marine engines, but which does not seem to have any advantage over the usual arrangement. The piston at III., instead of being tapped for lifting-bolts, has a hub threaded on the outside, so that a lifting-ring can be screwed upon it.

(f) **STRENGTH OF THE PISTON.**—Taking this piece in its simplest form—the single-disk type—it is possible to make a reasonably close calculation of the stress due to a certain steam pressure: but even here there must be simplifying assumptions which cause rational methods to be only approximate. With more complex, hollow pistons, only a rough guess at the stress can be made by calculation; so that empirical data, or the following of established good practice, is the chief resource of the designer. This is one of the numerous cases where each machine has

built is, in a sense, an experiment; therefore the development of good design has been largely a matter of evolution.

As always in the designing of a casting, there is here plenty of room for the exercise of good judgment. Thus it would be distinctly bad engineering to put a circle of core-holes in the face and of holes through the ribs at the same radial distance from the axis, especially if this radius is relatively so short that a large fraction of the metal in the face is removed. In this connection may be mentioned the occasional use of a sort of stay-bolt, made up of two core-plugs united by a shank, to tie together the faces of a hollow piston.

In rare cases the transverse face of the piston is not a surface of revolution: examples are described or illustrated in *Power* for Nov. 1899 and Feb. 1905, and in *Engineering* for Dec. 11, 1903. The first is a Corliss engine with the valves in the heads and projections upon the cylinder which fill the port-spaces; the second is a large blowing-engine, with piston-valves lying across the heads in the air-cylinder, and the piston conformed to the outer shape of the valve-casings. In such a design it is absolutely essential that the piston be secured against ever getting loose and turning on its axis.

(g) THE PISTON-ROD.—The chief point of interest in the construction of this simple piece is the manner of securing it to the piston and to the cross-head. At the piston, the most common fastening consists of a cone-fit and a nut. With a long taper, of 1 in 20 to 1 in 40, the piston is fitted against a shoulder, to limit the wedge-action of the rod, which would otherwise be likely to burst the piston. This shoulder may be formed by a reduction in diameter, as in Figs. 254, 256, and 277; but more usually, and better, there is a collar on the rod, as in Figs. 247, 251, 274 IV., 275 I., and 276. With a shorter cone the shoulder can very well be omitted, as in Figs. 267, 270, 274 I., etc.; although examples of a short cone in combination with a collar are given in Figs. 260, 265, and 279 II. In Figs. 6 and 281 III. are seen related designs with cone-fit and screw, all within the piston-hub. Parallel or cylindrical fits are sometimes used, as illustrated in Figs. 274 III., and 294; while Fig. 274 II. shows a simple screwed joint,

with a collar and a safety-rivet. The keyed joint, as in Fig. 27, was formerly quite common, but is now rarely used at the piston. The nut may be wholly sunk into the piston as in Fig. 277, partly as in Figs. 247, 256, 267, etc.; and may either be of the hexagon form or be slotted for a spanner-wrench.

At the cross-head, glancing forward to Figs. 282 to 297, we see that in stationary engines the rod is usually screwed into the head and secured by a jam-nut; though sometimes the threaded hub is split and clamped upon the rod, as in Figs. 290 and 297. In the locomotive the cotter fastening is used almost altogether, as seen in Figs. 286 to 289. With the marine cross-head, Figs. 293 to 296, radically different from the other types, a joint similar to that at the piston is usual.

*Strength of the Rod.*—The piston-rod is subjected to rapid alternating tension and compression, and must therefore be designed for a low working-stress. The compressive force, which the rod resists as a "column," with a tendency to yield by bending out of line, is the more severe in its effects; so that if the body of the rod is made strong enough for compression, a smaller cross-section can be used at the ends, in the fastenings, where simple tension is the predominant stress. But a sudden reduction of cross-section where the rod enters the piston is not a good feature, because any secondary bending action will tend to concentrate heavy stress at this point, and thus be likely to cause breakage. With the short cone and no collar, Fig. 267, etc., there is a very considerable enlargement at this point. But cases where a large enlargement of section is used even more avowedly for the purpose of getting increased strength against bending are seen in Figs. 256 and 289. In the former especially the maximum of stiffness is aimed at, because the general arrangement of the engine is such that at the cross-head the rods may have to resist very strong bending actions. They are in this case made hollow, so as to be as strong as possible for a given weight. These hollow rods, of nickel-steel and closed at the ends, are first forged with enlarged ends, drilled, and rough-turned on the body; then the ends are heated and hammered down, and the rod finished after whatever of oil-tempering and annealing is required.

The form of the piston-rod in tandem-compound engines is shown in Figs. 263, 279 II., and 438, with a reduced diameter for the part extending to the second piston. And the same arrangement is generally used when the rod is merely extended through the head for the sake of better guiding and support of the piston, as in Fig. 274 II. and III. and Fig. 279 II.; although sometimes, in horizontal engines, the tail-end is kept at full size, or the whole rod even enlarged and made hollow for greater stiffness, especially when a little carrying slide-block is placed under the outer end, as in Figs. 267 and 221.

(h) PISTON PACKING.—In the very earliest practice, a soft packing of hemp was wound around the piston-rim. With better facilities for machining the cylinder-wall, and especially with higher steam-pressure, metallic packing soon came into universal use. Almost always, this consists of flexible packing-rings, held in grooves in the piston and pressed outward either by their own elasticity or by separate springs behind them. The rings are generally made of hard, close-grained cast iron, although steel is sometimes employed and brass used to be. Occasionally the rings are made solid (uncut or with ends joined), and changed or adjusted as wear occurs. In connection with piston-rings, two matters are to be considered: first, the production and regulation of the requisite pressure between rings and cylinder-wall; second, the design of the ring-joint, which should not leave a gap for the passage of steam.

*Self-elastic Rings.*—Experiments made soon after the introduction of the plain snap-ring showed that a pressure of three or four pounds per square inch of contact-surface was enough to prevent leakage. The discussion of the elasticity of spring-rings and of the flexure necessary to a given pressure will not be taken up here; though some idea of practice can be got from sketches I., II., and III. in Fig. 280, where the free diameter of the ring is marked. Sometimes the ring is tapered toward the ends, as in Figs. 276 and at I.; more frequently, perhaps, it has a constant cross-section. In most cases the ring is turned to the larger diameter and pressed into the cylinder after a piece has been cut out; but a rather more uniform pressure can be got, as in Figs. 279 I.,

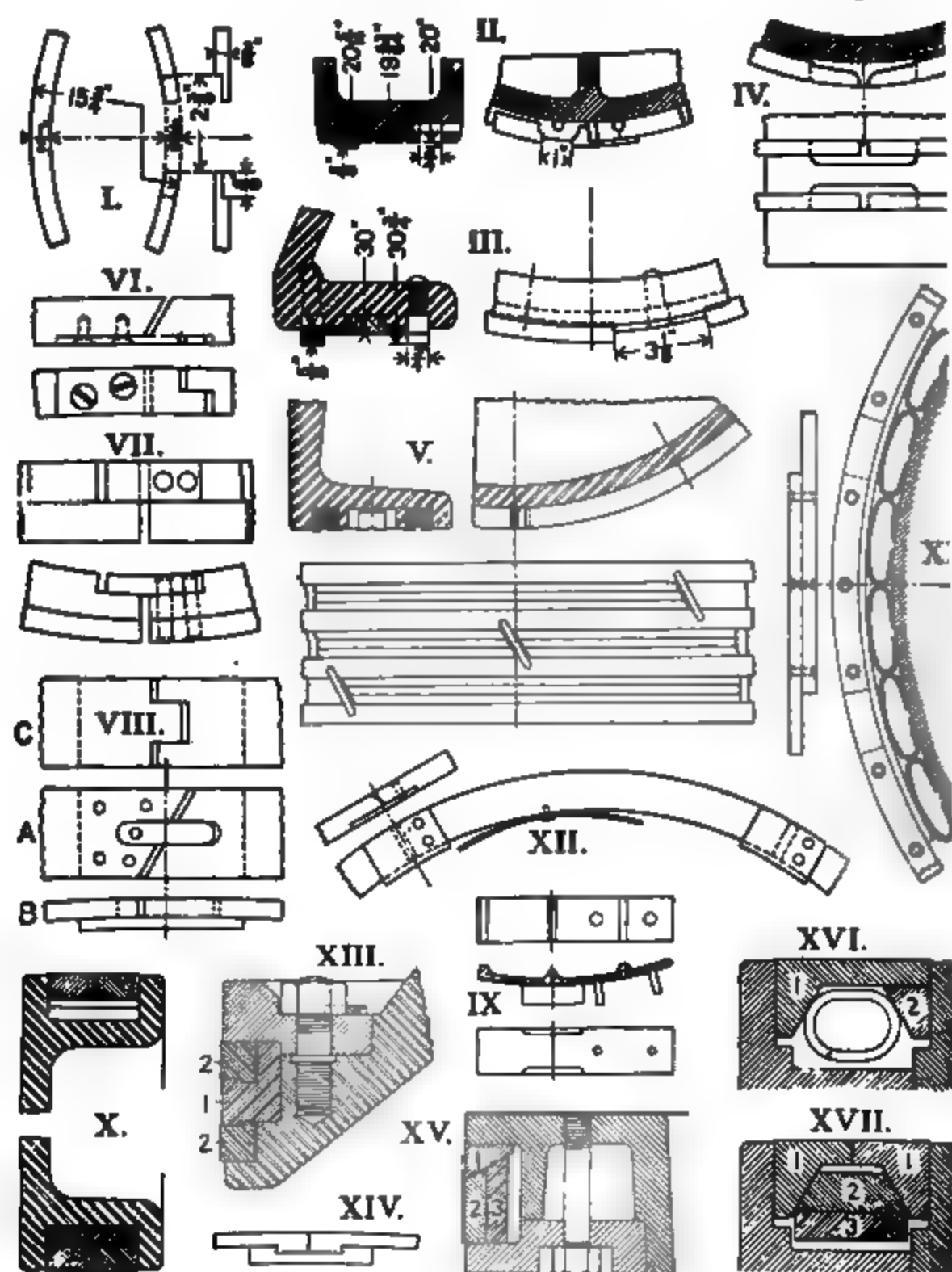


FIG 280.—Details of Piston-packing.

I. Ring for 15" piston, Fig. 251; II., III. Details from Figs. 247 and 250; IV. Ring-joint in Fig. 274 IV.; V. Detail from Fig. 274 II.; VI. Joint for ring in XIII.; VII. Detail from Fig. 279 I.; VIII. Joints for thin broad rings, for large pistons; IX. Ring-keeper from Fig. 279; X. Marine piston with steel spring-ring back of cast-iron rubbing rings; XI. Type of ring used in low-pressure piston, Fig. 279 II; XII. Ring for large vertical Corliss engine, Fig. 272; XIII. Restrained rings, British Navy practice; XIV. Type of self-restrained ring; XV., XVI. Wedge devices, for restraint of ring and to keep steam from getting under ring.

by cutting out a shorter piece than will let the ring close in to the diameter of the bore. then carefully clamping the ring to a face-plate and bringing it to size with a light finishing cut.

*Rings with Springs.*—In Figs. 278 and at X. we have examples of the continuous (one-piece) ring with spring backing: in this case the ring is turned just to the cylinder-diameter and sawed across. The joint at VIII. A is suitable for a ring of this type, while that at C would be made in a ring which depended partly upon its own elasticity. Rings in segments are illustrated in Figs. 277 and 280 XI. and XII.; this type of arrangement shows great variety as to the form of the segment and as to the kind of spring used. Always, provision must be made for holding the springs in place and keeping them properly spaced. In Fig. 277, each keeper has upon it a little stud which projects into the center of the helical spring; in Figs. 278 and 280 XI. the springs fill the whole annular space; in XII. the spring is intentionally fastened off the middle of the segment, toward the end where the keeper is riveted fast. Another type of spring is seen at XVI.; and the continuous corrugated ribbon, or wave-spring, as in Fig. 281 II. C, is sometimes used.

*Grooves in the Contact-surfaces.*—When a fluid is flowing through a long narrow channel, the current is effectively broken up and its velocity greatly checked by occasional abrupt enlargements of the channel. This principle can be applied by cutting small grooves in the piston-face, as at II. and XIII., or even in the surface of the ring, as at V.; and another application is seen in Fig. 281 IV.

*Ring-pressure Due to the Steam.*—The pressure actually existent between packing-ring and cylinder-wall really depends, in most cases, not so much upon spring-action as upon the pressure of the steam. The ideal condition is to have the ring fit its slot very closely, being free to move, but not letting steam leak into the space behind it. This kind of leakage is very hard to prevent; so that, during a part of the stroke at least, the ring is likely to be very strongly pressed outward, with undue friction and wear as the result. In some lines of practice, holes have even been drilled through the side of the piston into the lowest part of the slot, to

let the steam get in and push the ring out. At the opposite extreme of practice lie the various devices for restraining the rings and for more effectively keeping the steam out of the slot, as illustrated in Fig. 280 XIII. to XVII.

*Restrained Rings.*—At XIII. is seen a design in which the packing-rings 2, 2, are formed with a shoulder, and restrained from undue expansion by collars on the solid steel ring 1—the whole system being removable without disturbing the piston-rod connections. In this case, new rings would be given just enough play to let them wear to a good surface, after which they would be practically solid rings. The idea of a self-restrained ring is represented by the sketch at XIV.: this type of coupling has been used as also the oval link, like that in Fig. 326 II. Fig. 280 XV. is typical of several patented designs; ring 1 is solid, 2 is a rubbing ring without spring, 3 is a strong spring-ring. By proper adjustment at the start, ring 2 can be left free just to wear to a good fit in the cylinder; whereupon, as is claimed, the slight amount of wear which takes place on the side surfaces in the slot will allow enough expansion to keep the piston tight, the slant between the rings being accommodated to different conditions of service. This slant also serves to wedge the rings fast in the slot and prevent side leakage.

*Side-wedge Action.*—This idea of keeping the steam from getting behind the ring has led to a number of designs, which are typified by XVI. and XVII., both examples being intended for vertical engines with extended piston-rods. At XVI. the spring is a helix of elliptical cross-profile, coiled on a straight axis, then bent around the cylinder; at XVII. there is a cast-iron wedge-ring 2 and a steel spring-ring 3. In either case, the pressure upon the oblique surfaces has a larger component to force the halves of the rubbing ring apart, against the sides of the slot, than to force the whole ring outward. It is said that these arrangements are quite effective in diminishing friction and wear; they are all of European design.

*Ring-joints.*—Turning now to the ring-joint, and running over the examples given in Fig. 280, we have the simple butt-joint II. and III., with provision made for holding the two joints near the bottom of the piston, where its weight is depended upon to hold it down and prevent steam from getting to the gaps in the



rings. The arrangement at V. is essentially the same, except in the use of little stop-plates, set into slots which are partly milled out, then finished by hand. In IV., as also at IX. and XII. and in Fig. 277, we see the use of little brass keepers to form the joint: these are surest to close the gap when riveted to one of the ring-ends, or when right over the springs; so that IV. depends upon the weight of the piston, as do all the other locomotive designs shown. In general, these keepers serve to prevent leakage and to make the ends match; and both these functions are taken by the little side plate in VI. Where two rings are together in one slot, they can always be placed so that the joints are far apart, and no lap at the joint is then needed: examples of this arrangement are seen at VII. and X. In VIII., the cover-plate fastened inside the joint is called a "palm"; and with it some form of tongue or lap is used to bridge the gap.

(i) PISTON-ROD PACKING.—For use in stuffing-boxes, around piston- and valve-rods, soft or fibrous packing was long the only material employed; and it is only in comparatively recent times and under the more severe conditions of service and pressure that metallic packing has been coming into general use. Formerly, loose strands or braids of hemp soaked in tallow were used; now there is a host of ready-made packings on the market, in uniform and graded sizes so as to fit and fill the stuffing-box neatly. These are made of vegetable fibre, asbestos, or rubber, in various combinations, with graphite frequently incorporated as a lubricant.

*The Stuffing-box.*—Examples of the common stuffing-box have been given in Figs. 6, 250, 251, 253, 262, 263, 267, 269, etc. Always there is an annular space around the rod, closed by a flanged bushing called the "gland," which can be screwed down to compress the packing. Sometimes the confining end-surfaces are flat, as in Fig. 269; oftener, one or both are made conical, so as to force the packing more closely against the rod. To allow for initial or acquired faults in alignment and for wear of the sliding surfaces of piston and cross-head, the rod must fit but loosely where it passes through the cylinder-head and the gland. Very often bushings of brass or linings of babbitt-metal are used, as can be seen in the examples just cited. Of course, the clearance must

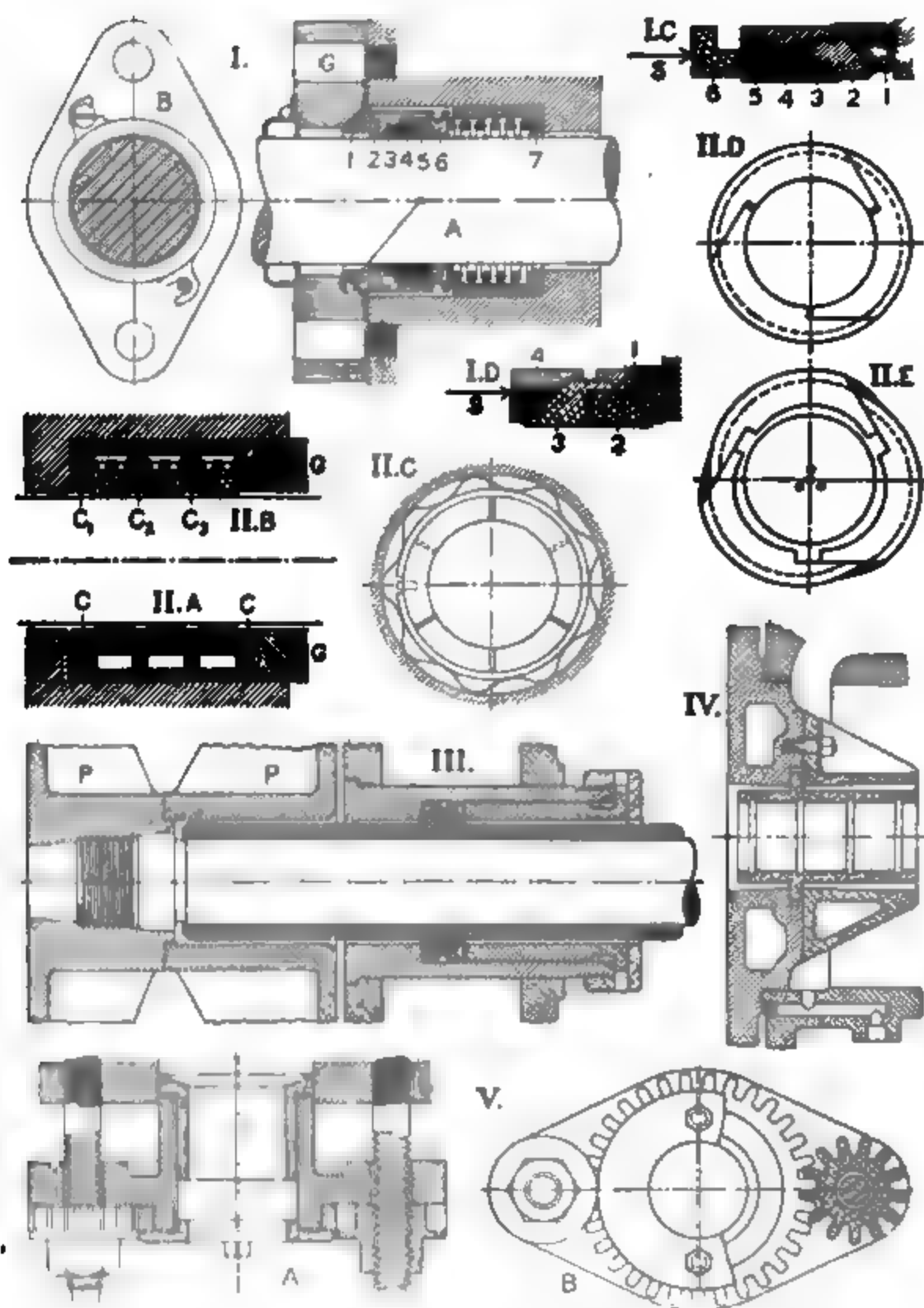


FIG. 281. — Types of Piston-rod Packing. I. "United States" metallic packing; II. Various ring-packings; III., IV. The plain solid bushing; V. Gland with geared nuts.

not be wide enough for the packing to squeeze into it. A very simple and effective device is to use brass rings as in Fig. 269, making them fit the rod quite closely, but with as much clearance at their outer circumference as the rod has in the head and gland.

*Metallic Ring-packing.*—The metallic packing belonging to Fig. 247 is drawn to a larger scale in Fig. 281 I. The gland G is here a mere heavy cover-plate, made tight by a copper-wire ring-gasket. The ball-ring 1 is seated upon the gland with a spherical ground joint; the casing 2 is free to move sidewise, and contains the three babbitt-metal packing-rings 3, 4, 5, which are the only parts that touch the rod. These, as pieces requiring comparatively frequent renewal, are made in segments so that they can be put in without disturbing the rod-connections, and are placed in the cup so as to “break joints.” The follower 6 is pushed up by the spring, which is itself held in the light casing 7. This spring is not expected to do much more than hold the rings in place: it is the steam-pressure that wedges the rings into the cup and makes the joint tight, so that the tightness of the joint varies with the pressure of the steam to be held.

The same packing, with different proportions, is shown in detail at I. C—the difference consisting in the slant of the confining surfaces of the cup, which causes all the rings to come into action, although the first one still takes most of the wear. And at I. D is given the detail of another packing of this same wedge-action type, with brass rubbing-rings.

*Collar-rings.*—Another type of ring-arrangement is represented by the two examples given in II.: the rings have parallel faces and are held between the collars of a cast-iron casing, and the segments are pressed inward upon the rod by peripheral springs. In the design shown at A and C, each ring is divided radially into three segments, and the two rings in one compartment break joint, being kept in place by little dowels which project into the gap in the elastic confining-ring. With this arrangement, the casing must be made in halves, divided lengthwise; and as the figure is drawn there is hardly room for the needful clamping device to hold the halves together. At B, D, and E the springs are helical, and form elastic bands around the rings. The form of the seg-

ments and the manner in which the ring can open and close shown at D and E. These ring-packings are made in a great variety as to form and arrangement. Very often there are several complete sets of rings, the space between them being used for lubrication or for the draining away of water. On vertical engines especially there is a strong tendency to leak water around the piston-rod. In *Power* for April 1902 is given a full description of a great number of packings of this general type.

*The Solid Bushing.*—Another form of metallic packing, the plain solid bushing, is shown at III. and IV. The first is from SWEET's "Straight Line" engine, and has a high degree of accommodation to any irregular movement of the rod. The long, close sliding fit between the rod and the babbitt sleeve effectively prevents leakage. In this case the piston P is made in halves, one of these is recessed to make room for the bushing. The arrangement at IV. is used between the cylinders of a close-connected tandem-compound locomotive—see Fig. 438. The bushing of bronze or hard brass, can adjust itself sidewise: and in arrangements of this sort, long used in pumps like that in Fig. 228, wear has been found to be almost imperceptible.

The materials used for the rubbing surfaces of packings of several types shown are, hard babbitt metal, brass or bronze and cast iron: they must be all of such grade and finish that they will not cut or score the rod.

The last drawing in Fig. 281, at V., shows an arrangement sometimes used on marine engines with fibrous packing: two nuts are connected by gearing, with assurance that the glands will always be kept "square" with the rod.

(j) THE CROSS-HEAD.—The three typical forms of this part are well represented by the next three figures. In Fig. 282 is shown a four-bar cross-head (to be held by four guide-bars, the wing type); and from the way that the two slide-blocks project sideways we shall call this the wing type. It can be seen in place in Fig. 283, II. and III. and Fig. 216; but it is far less common than formerly now that enclosed guides are so generally used. The most prevalent form, the block or trunk type, is represented by Fig. 284, the essential feature being that the guided surfaces are above and

below the cross-head, central in the plane of the engine mechanism. The one-sided slipper cross-head is shown in Fig. 284, here intended for a non-reversing engine, and with comparatively little bearing-surface on the top of the sole-plate. The fourth example, given in Fig. 285, is a modification of the last type, the cross-head surrounding the guide-bar, instead of being partly enclosed by the guiding surfaces.

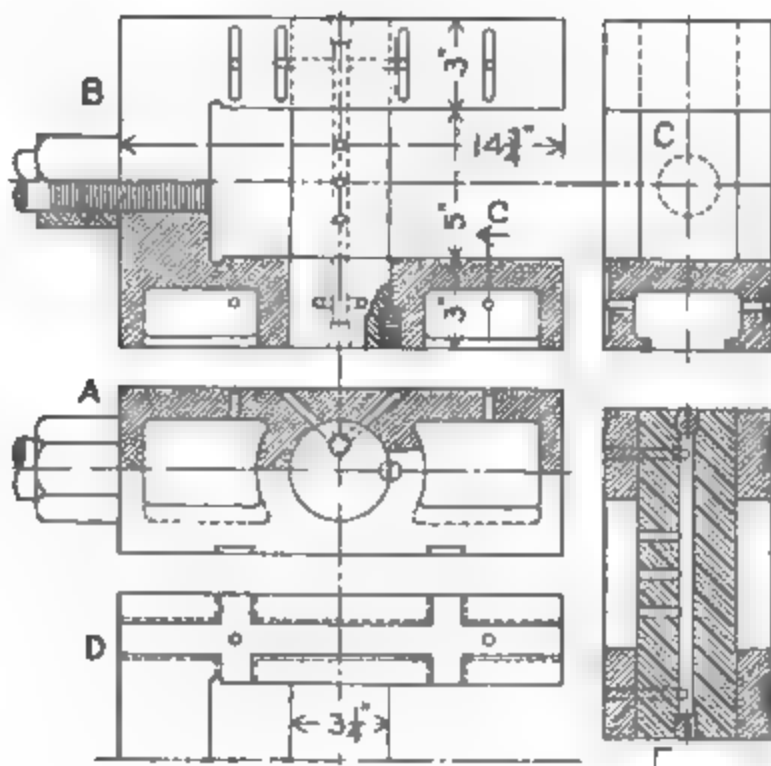


FIG. 282.—Four-bar or "Wing" Cross-head, to fit guides shown in Figs. 237 and 298. Scale 1 to 10.

In general we note that any cross-head has the two functions of joining the connecting-rod to the piston-rod and of sliding upon or between the guides: it consists therefore of the body, which begins with the hub for the piston-rod and is usually forked to receive the wrist-pin; together with the sliding surfaces or shoes, of whatever form. Except in locomotives, the piston-rod is nearly always screwed into the hub and clamped by a jam-nut or equivalent device. As to the wrist-pin, we have here two examples of cylindrical fits, two of taper fits. In Fig. 282, for a connecting-rod with strap end, the pin is made with a driving fit and held by a screw-key; in all the other examples, the pin is easily removable, as it must be taken out in order to break connections, with a solid-end rod. Usually, the wrist-pin is made of machinery steel, often case-hardened and ground; but in Fig. 284 it is of hard cast iron, hollow for lightness, and arranged so that it can be turned through a quadrant occasionally, to equalize wear.

In Fig. 282 the whole piece is of cast iron, and the principal,

bottom rubbing surface is composite, the grooves (most of

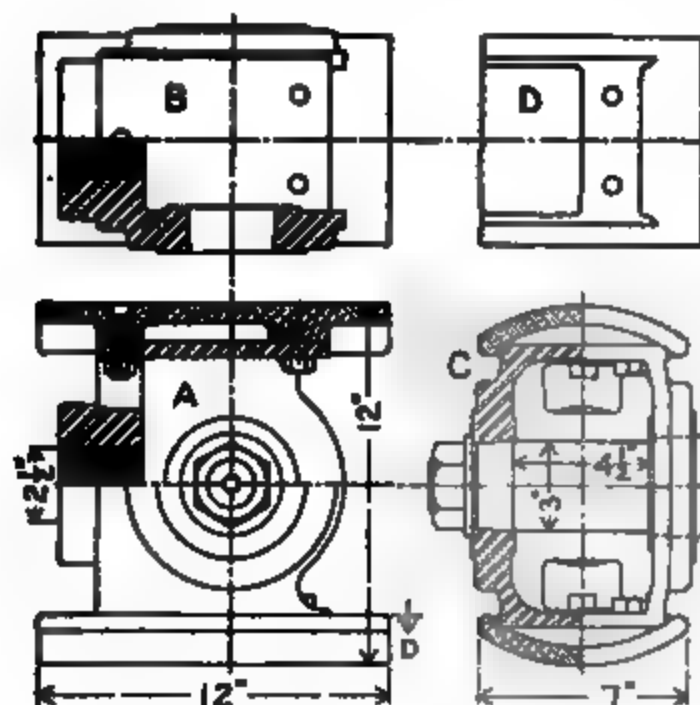


FIG. 283.—Cross-head of "Block" Type, for Fig. 286; compare Fig. 268. Scale 1 to 10.

shown at D) being fitted with babbitt-metal. 283 has a cast-steel block and brass shoes. In Figs. 3 and 4 the cross-heads are all of cast-iron; here, in Fig. 284, the body is cast steel and the bars are cast iron or brass preferred. Fig. 285 shows a cast-iron pattern, babbitt facing. The lubrication arrangements are fully shown on Fig. 282, consisting of grooves in the upper surface which gather oil from the top guide-bar and feed it to the wrist-pin, as shown at E, to the lower slide. In some of the figures which follow certain

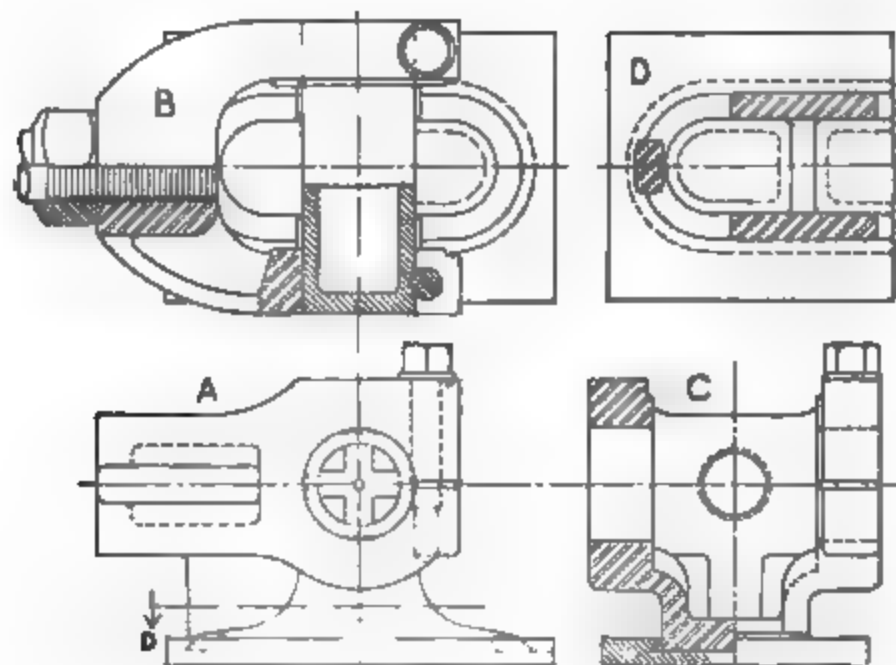


FIG. 284.—Slipper Cross-head, detail from Figs. 3 and 4.

details of this sort are omitted, because they would unduly complicate the drawings.

(k) LOCOMOTIVE CROSS-HEADS.—For this class of engines, by far the most common form of cross-head is the modification of the block type shown in Figs. 286 and 287. The first example is a good standard design, very generally used; the material is cast steel, for both body and shoes; and the latter are lined with a coating of pure tin, which is put on like a solder, adhering of itself to an acid-cleaned surface. The top view D is intended to bring out most clearly the strength of the fork; E is a view of the shoe from the inside, showing the recesses which receive the flanges of the body; while F is a detail of the piston-rod joint. In the second design,

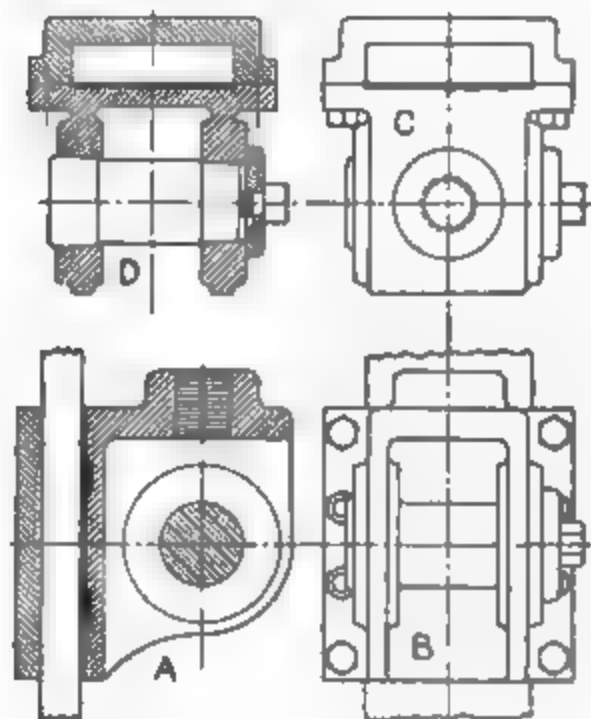


FIG. 285. — Slipper Cross-head "Inverted," surrounding the guide-bar, from Fig. 202.

reduction of weight is made a prime object; and advantage is taken of the fact that the lower surface, which presses upon its

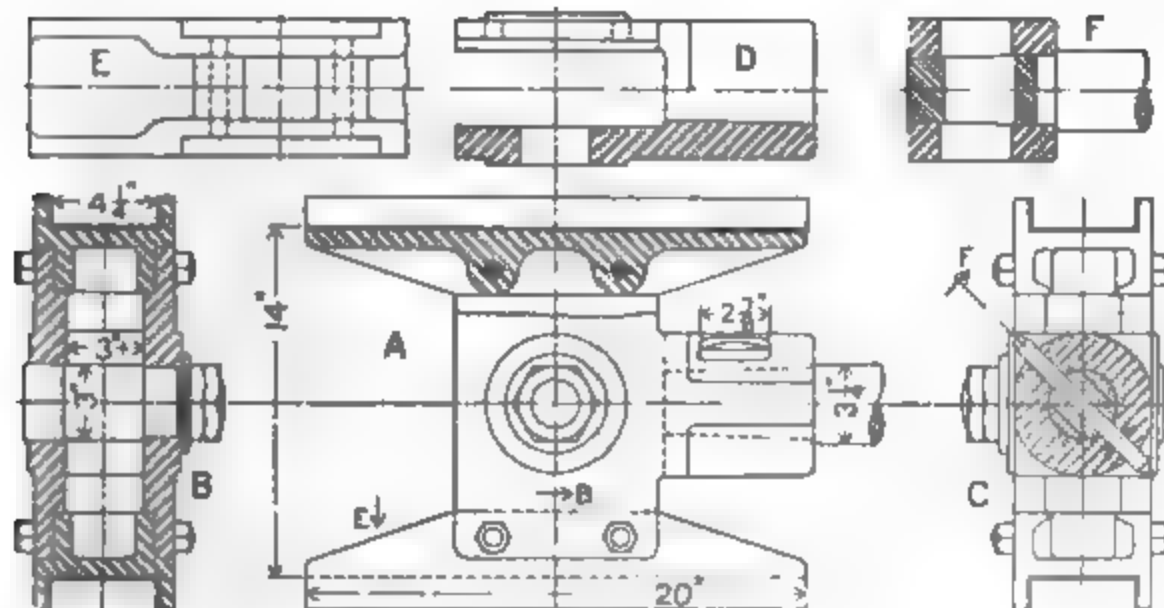


FIG. 286. — Standard "Alligator" Cross-head, for 20' by 24' locomotive, Fig. 247. Scale 1 to 12.

guide-bar only in reversed running of the engine, is very used, and can therefore be made much smaller without danger of excessive wear. Further, in order to shorten the guide-bar at the back end, the shoes are set forward from the center-line of the wrist-pin. These shoes are of brass, which is likewise often used in designs like Fig. 286. On general principles, it is better to have the pin central in the slide-block, for only thus can a uni-

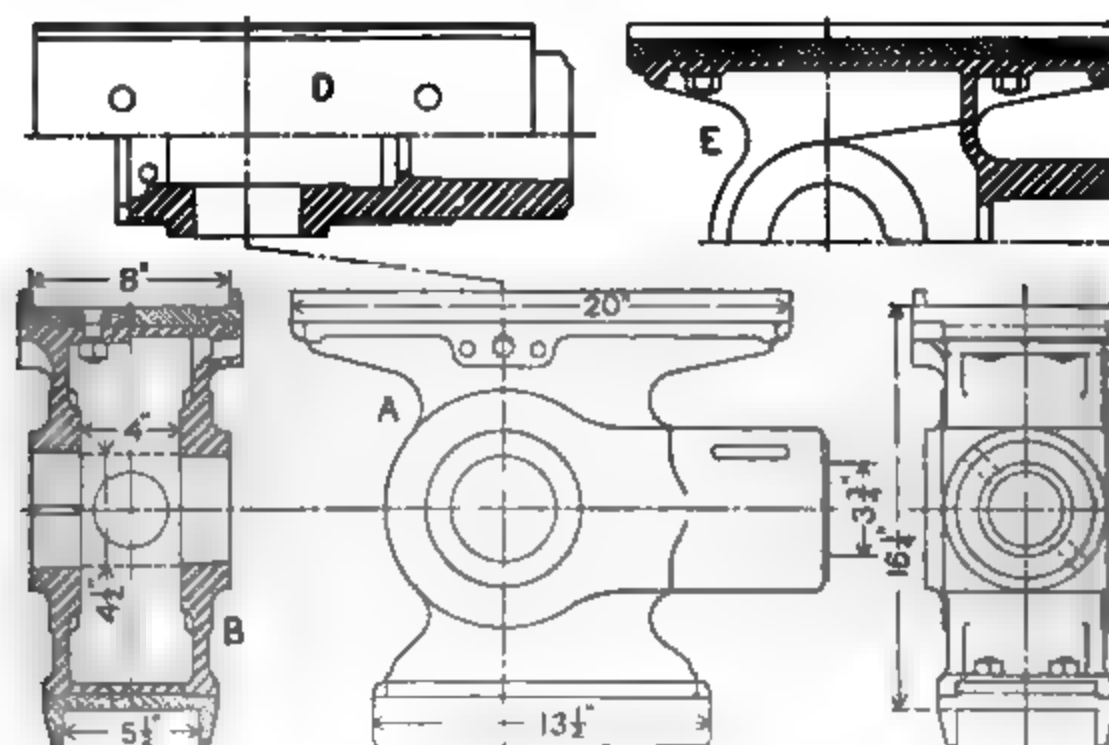


FIG 287.—Special Design Alligator Cross-head, for 19'' by 26'' fast-run locomotive. Scale 1 to 12.

distribution of the bearing-pressure be secured; but a moderate amount of eccentricity does no great harm, and is not infrequently seen, especially in marine engines. Cross-heads are proportioned for a maximum bearing-pressure, near the middle of the stroke of from 40 to 60 lbs. per sq. in. upon the guide-bars.

*Slipper-type Cross-heads.*—The usual form of the slipper cross-head for the locomotive is shown in Fig. 288, where the top shoe, made hollow to save weight, works between two bars and is held to the body by tightly fitted through-bolts. The top shoe exerts the guiding-force in former running; the lower one serves to keep the cross-head from moving out of line sideways, the flanges which unite the shoe to the body of the cross-head being li-



with thin brass plates, riveted fast. The main bearing-surface, on the top of the shoe, is composite; shallow pockets are drilled in the brass and filled with babbitt metal.

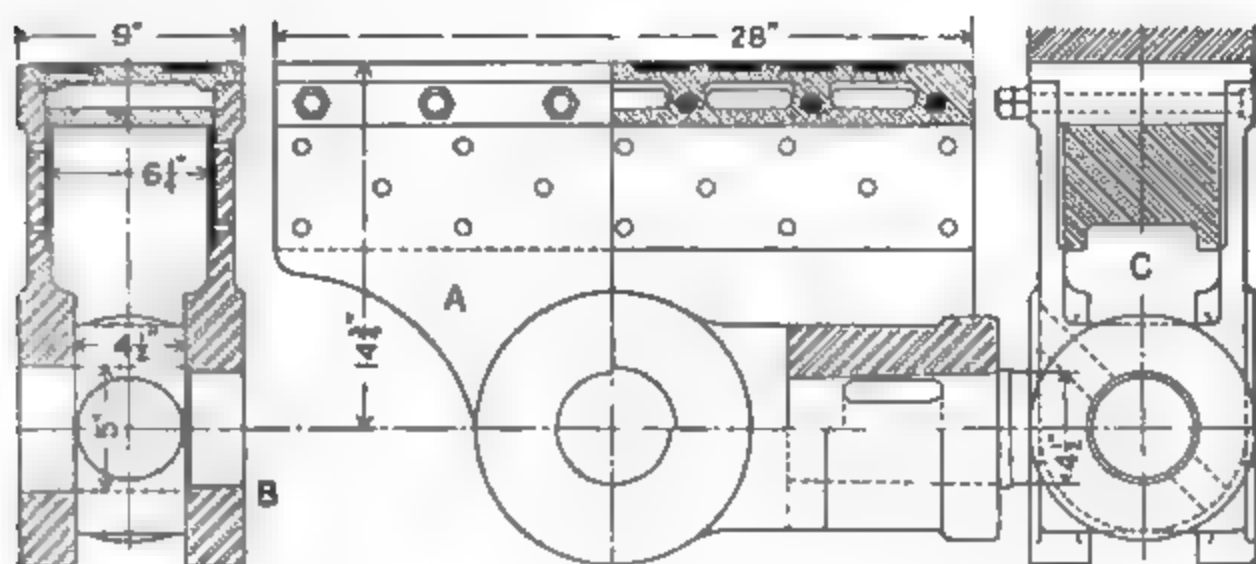


FIG. 288.—Slipper-type Cross-head, for extra big freight locomotive, 19" and 32" by 32" tandem-compound. Scale 1 to 12.

The design in Fig. 289 is rather out of the usual line, approaching the marine type of slipper cross-head—compare Figs. 295 and

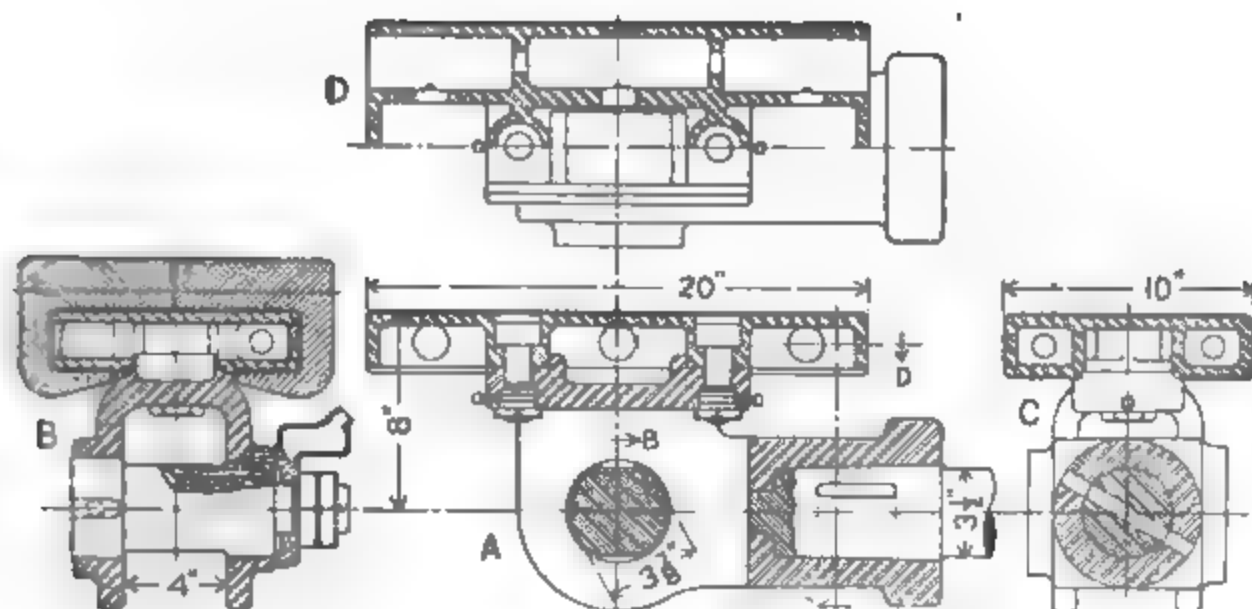


FIG. 289.—Locomotive Cross-head of the Marine Type, Pennsylvania Railroad express engine, 20 1/2" by 26". Scale 1 to 12.

299. The broad, thick shoe or sole-plate is a light steel casting, coated with tin about 1/16 in. thick. The bolts must transmit the

full guiding-force in reversed running, and are therefore made of ample size.

*Piston-rod Connection.*—A notable feature of all these cross-heads is the fact that the piston-rod is held by a cotter, chiefly, it appears, because this form of joint can be most easily taken apart, while offering a high security against shaking loose. There are two typical arrangements of the joint; in Figs. 286 and 288,

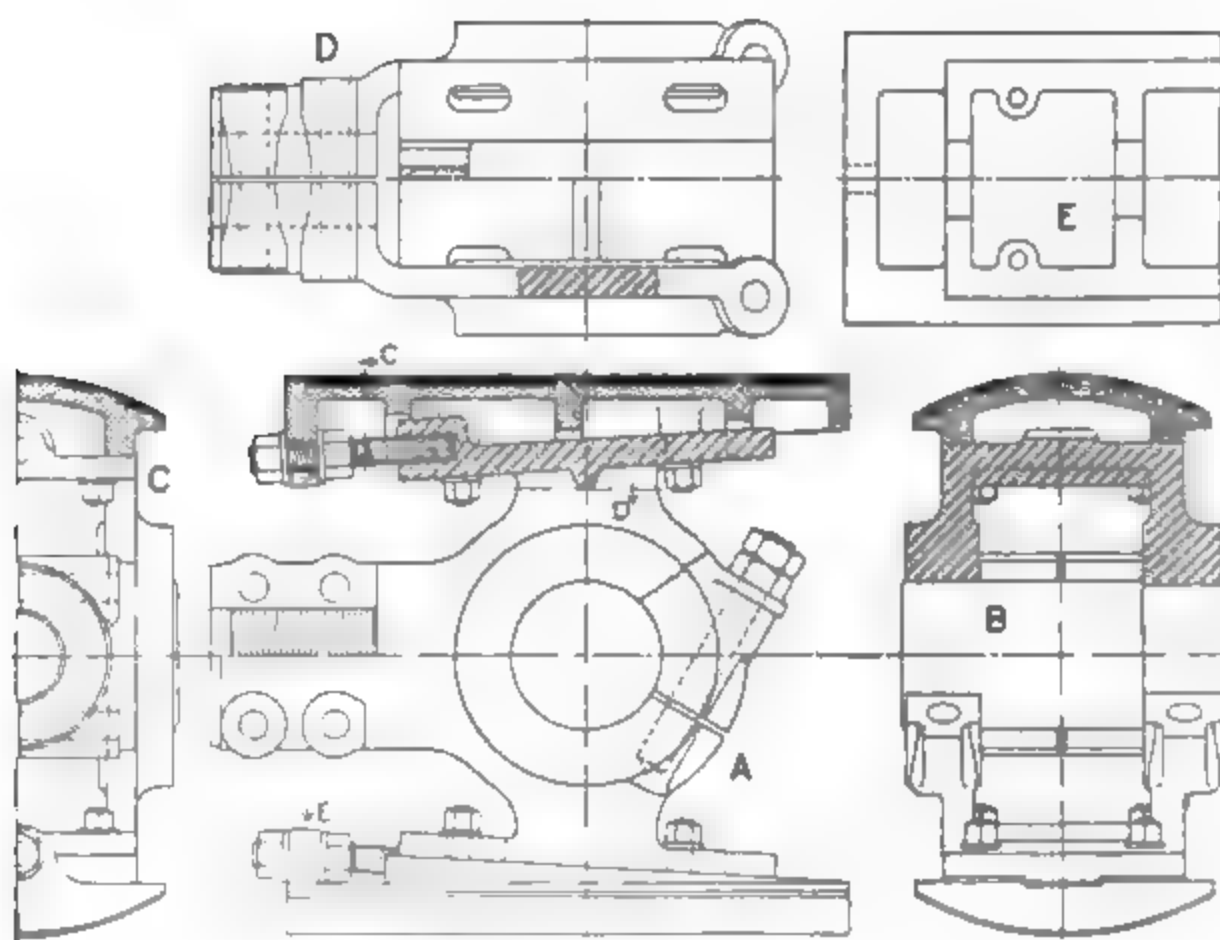


FIG. 290.—Cast-steel Cross-head, vertical Buckeye engine.

the entrance of the cone into the hub is limited by the shoulder on the rod; in Fig. 289 the end of the rod seats itself upon the bottom of the hole. In the first case, driving the key in hard will develop a heavy initial tension in the rod, most severe where the cross-section is reduced by cutting out the key-slot. In the second case, the key simply compresses the tip of the rod, and only the tension due to the steam-pressure is felt by the metal in the rod. Careful work is needed, of course, to insure that cone-fit and shoulder-fit shall coincide in tightness.

(d) **CROSS-HEADS FOR BORED GUIDES.**—This is the most common type for stationary engines, as already stated; and except in small engines there is some form of wedge-arrangement for adjusting the shoes. Fig. 290 has a broad bearing upon the body for the shoe, which is adjusted by an inclined screw and then strongly clamped by four stud-bolts. The inclined supporting surface is much smaller on Fig. 291; and the clamping-bolts are relatively

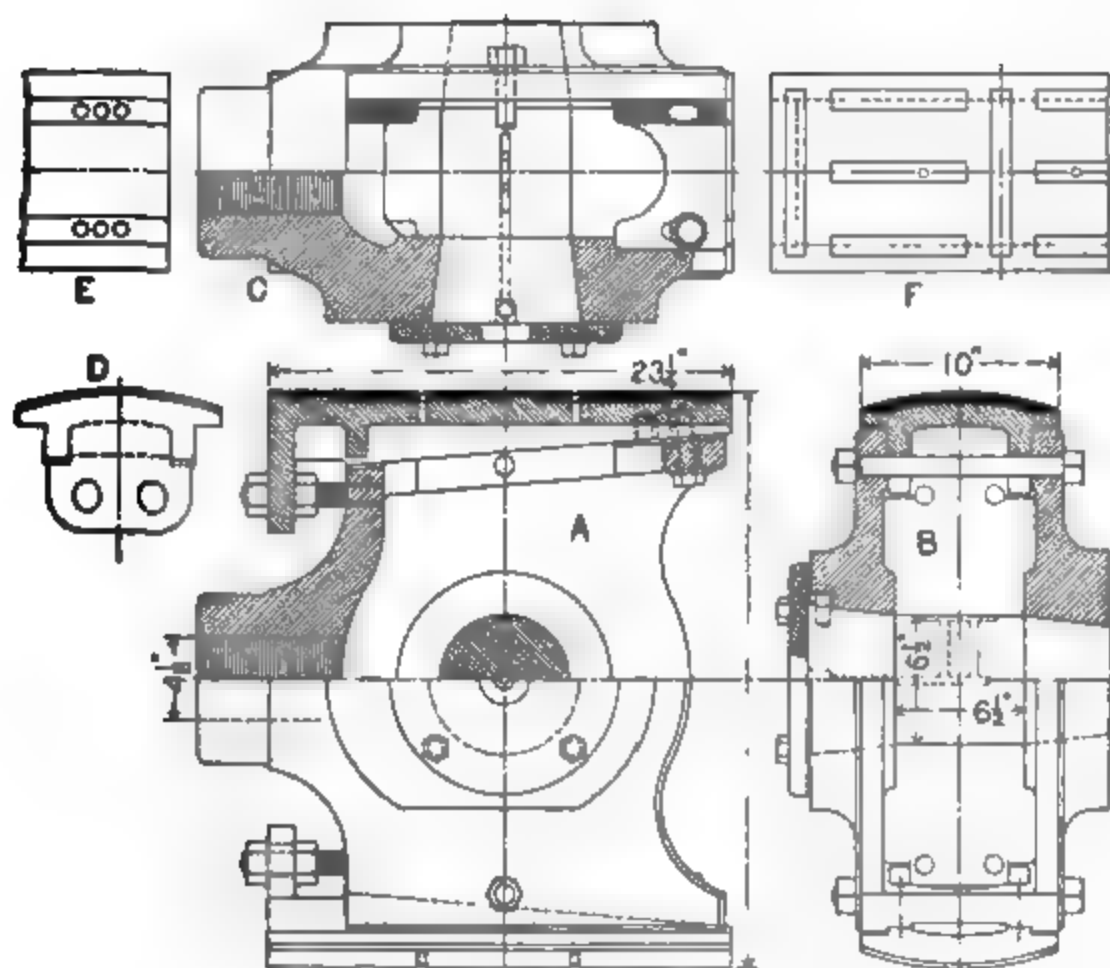


FIG. 291.—Cross-head for Corliss Engine, to go with piston in Fig. 278.  
Scale 1 to 15.

less strong, so that more dependence is placed upon the adjusting-screws for driving the shoe back and forth with the cross-head. In Fig. 292 there is a separate wedge between body and shoe, and the latter is held in place by its own end-flanges and has no endwise movement in adjustment. These three arrangements represent the typical variations in the great number of designs that have been gotten out.

The body of the cross-head in Fig. 290 is of cast steel, with weight of metal reduced to a minimum, and the shoes are of cast iron. Note the clamping-devices for gripping both piston-rod and wrist-pin. Figs. 291 and 292 are both designs for cast iron, but they can easily be enlarged and reduced in relative thickness when steel is to be used, for bigger and faster engines. A special feature of Fig. 291 is the way that the wrist-pin is held in place.

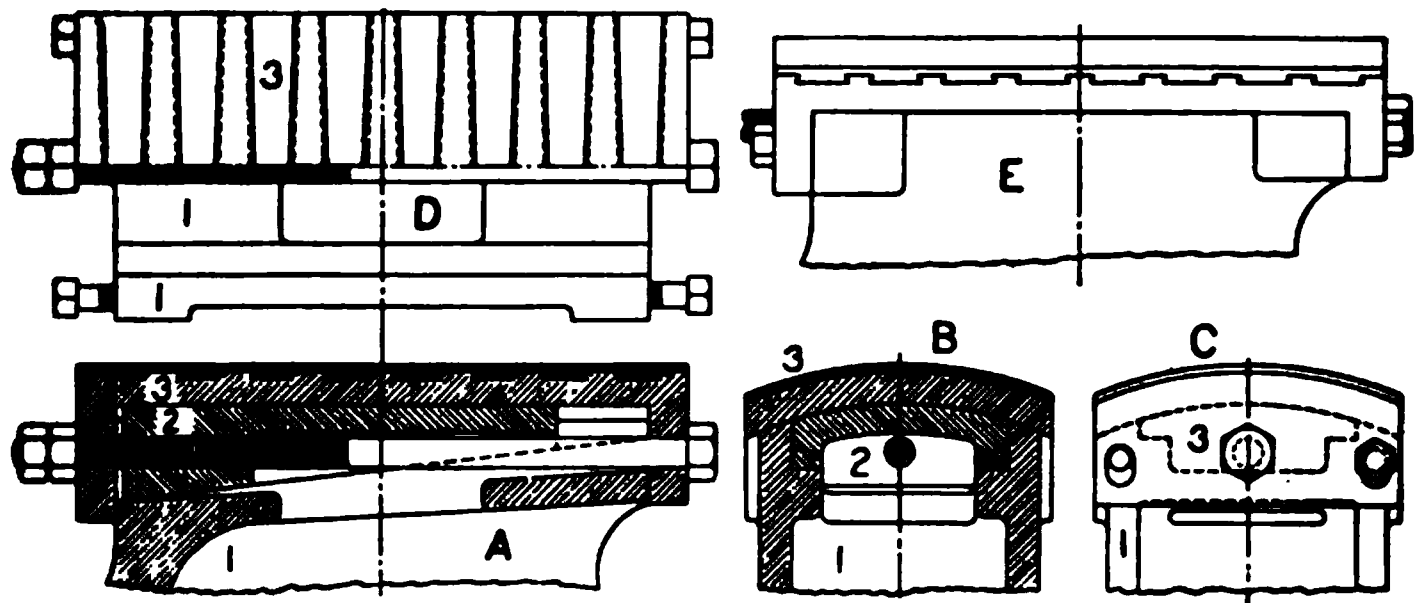


FIG. 292.—Corliss-engine Cross-head with separate wedges.

The manner of securing the facing of anti-friction metal is shown in detail in each of these examples. In general, the alternate dovetails on shoe and babbitt-shell are narrow, so that the latter is held fast in small portions. In Figs. 290 and 292 there is an approximate equality between the amounts of thin shell over the ridges and of thick shell in the grooves of the shoe. Fig. 291 shows the thin shell predominating, while the opposite proportion is seen on Fig. 295. On Fig. 292 the dovetails are tapered, so that the shrinkage of the whole shell in cooling will draw them tight, and compensate for their own shrinkage—for even though babbitt metal expands in solidifying, it must contract somewhat in cooling down from the melting-point.

(m) CROSS-HEADS FOR MARINE ENGINES.—Our three typical forms of cross-head, as adapted to marine-engine conditions, are exemplified in Figs. 293 to 295. Practically without exception, it is here the connecting-rod that is forked, not the cross-head; therefore the body of the latter becomes a plain block, through which the end of the piston-rod passes; and the two pins are i

the form of gudgeons, like trunnions, on the sides of this block. Then the four-bar cross-head takes the shape in Fig. 293, an engine

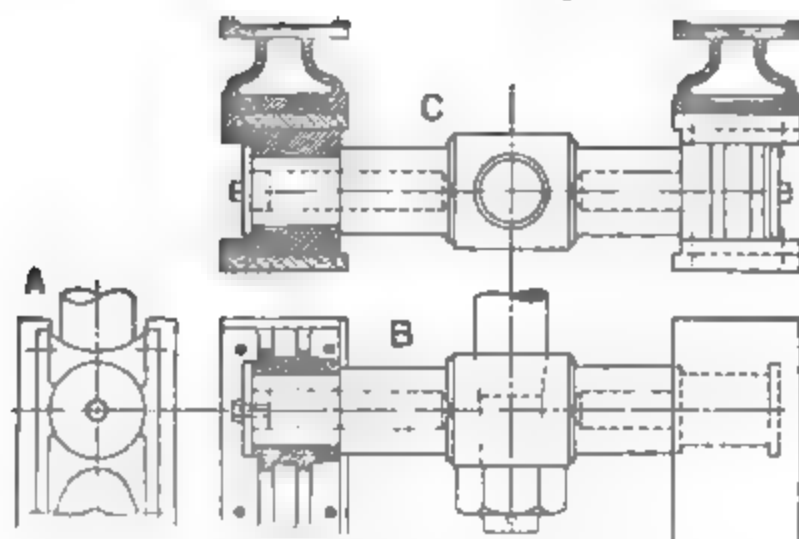


FIG. 293.—Marine Cross-head of the Four-bar Type, Dutch cruiser with cylinders 38", 49", 74" by 39". Scale 1 to 24.

with this arrangement being shown in Fig. 233. In Fig. 294 the shoes are cast pieces, bolted to the block; this type is generally used when the engine has A-frames, as in Fig. 232 (although that

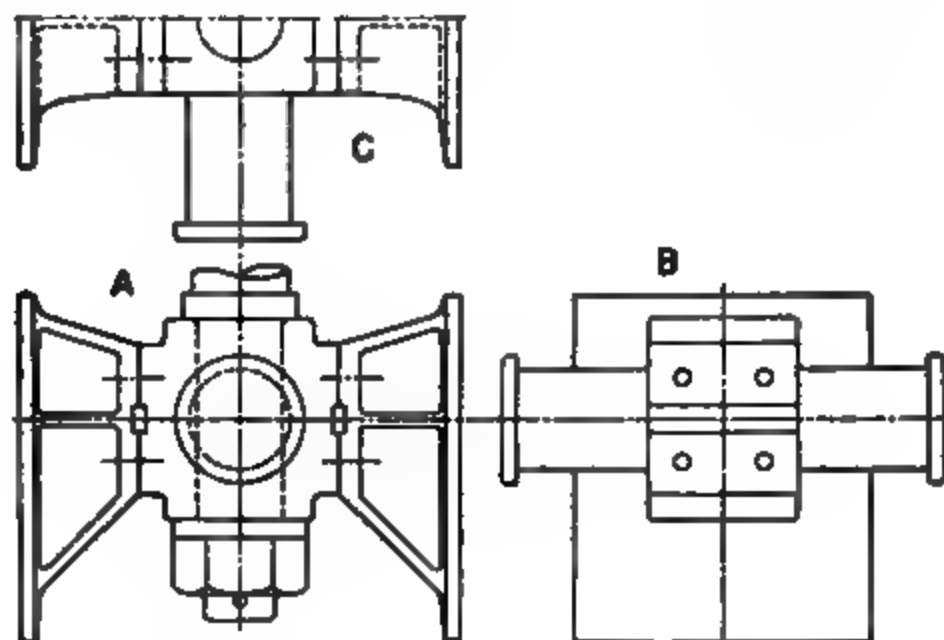


FIG. 294.—Marine Cross-head of the Block Type, French battleship, cylinders 44.5", 67", 105.5" by 48.2". Scale 1 to 24.

engine has slipper cross-heads). A good example of the third type, now most prevalent, is given in Fig. 295; here the slipper 2 is of forged steel, as well as the block 1, and the web can be made so thin

that the surface of the backing or top guides (see Fig. 299) is not much less than that of the forward or bottom guide. As it happens, these examples are all from warship engines, where it is of first importance that the total height of the machine be kept down and all have the rubbing-surface central below the wrist-pin. In merchant ships, however, the slide is usually made symmetrical.

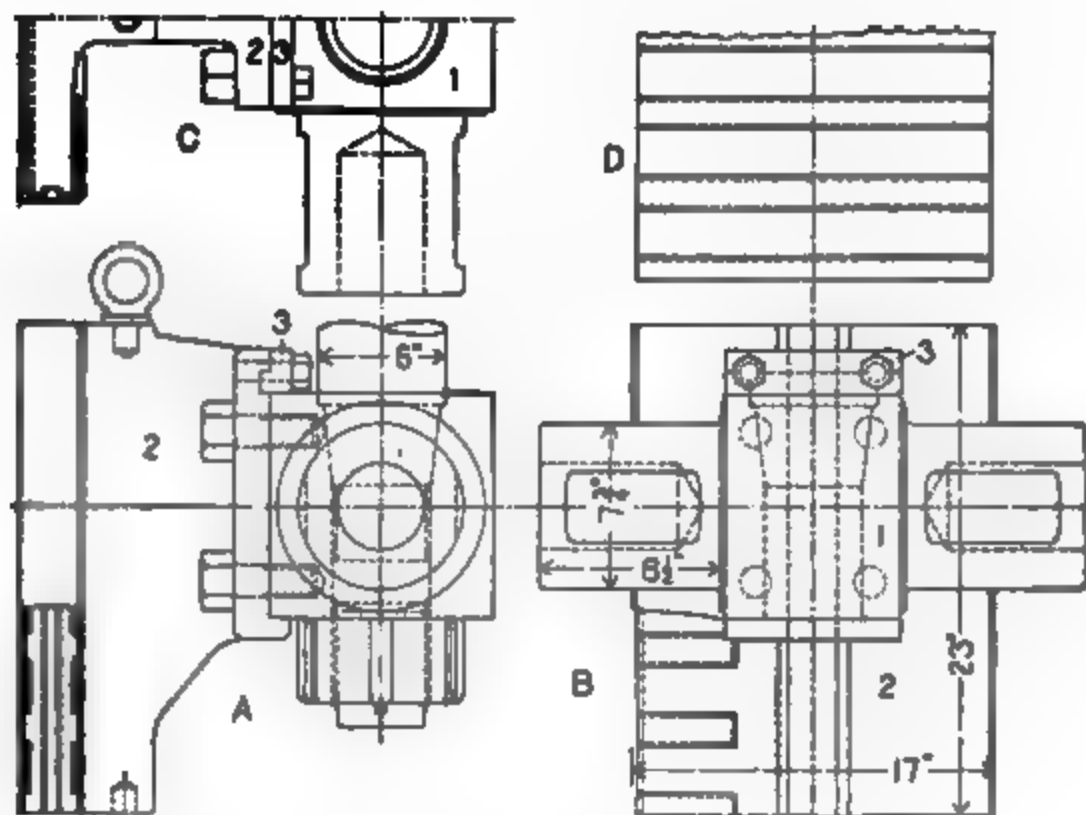


FIG. 295.—Slipper Cross-head, U. S. Battleship "Mississippi," 25½", 42", 69" by 48" (four engines). Scale 1 to 14.

*The Wrist-pin Fast in the Rod.*—Many marine engines of the smaller sizes have the wrist-pin joint made as in Fig. 296, where the pin is held in the forked connecting-rod, and the cross-head, which is forged in one piece with the piston-rod, carries the bearing. This particular example is from a small stationary, non-reversible engine, designed along marine lines; therefore very little holding down surface is needed on the top guides.

The unique design in Fig. 297 fills what would otherwise be a gap in the list of typical arrangements, supplying a solitary example of the combination of a forked cross-head with a wrist-pin fast in the connecting-rod. The cross-head as a whole is a very light steel casting, and the lower half of the hub, shown at D, is a separate

piece, strongly bolted fast and arranged so as to grip the piston-rod. The wedge-adjustment for the bearings is intended to accommodate itself to equality of pressure on the two ends of the pin. This very interesting design is rather at a disadvantage, however, when it comes into commercial competition with less expensive arrangements.

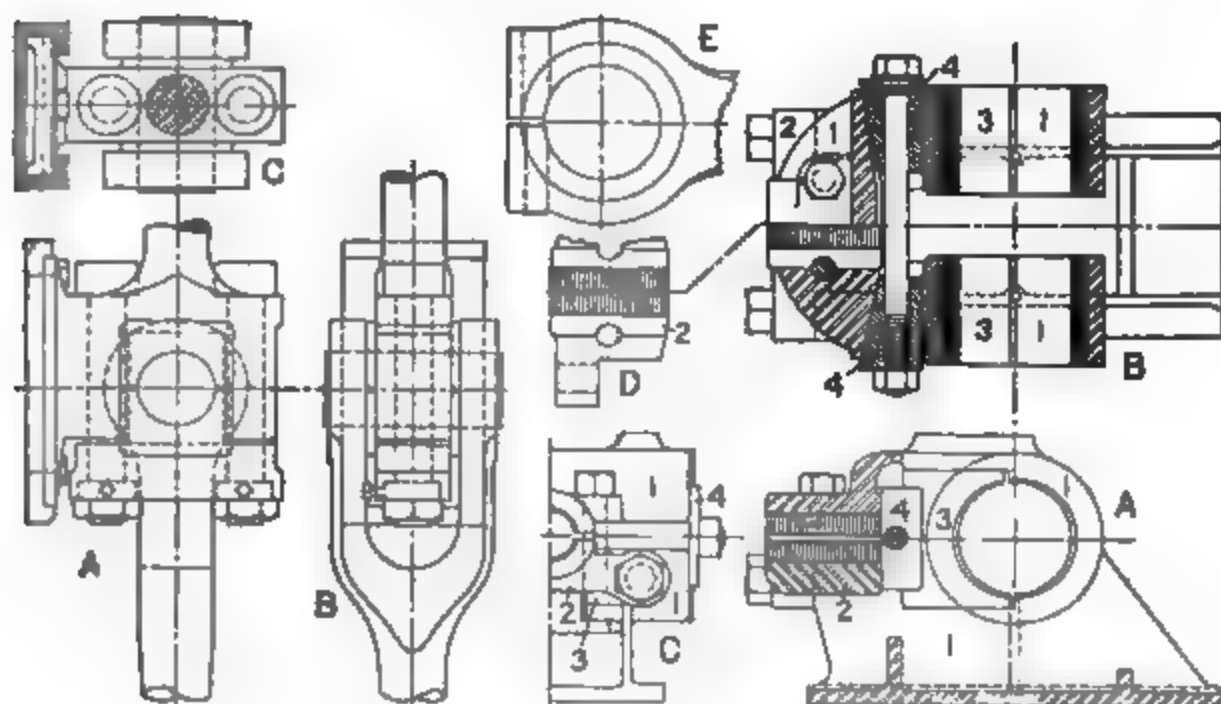


FIG. 296.—Wrist-pin Fast in Forked End of the Connecting-rod.

FIG. 297.—Cross-head of the "Straight Line" Engine.

(n) THE WRIST-PIN may be considered with regard to the shape and dimensions of its journal-surface, and to the manner in which it is held in the cross-head. In Figs. 282 and 283 we see pins on which the length of the bearing-surface is about one and one-half times the diameter; but in most cases these dimensions are nearly equal, or the journal is nearly "square." Sometimes the practically useless side surface of the pin (on top and bottom in a horizontal engine) is cut away, as in Figs. 289 and 295. The movement of the rubbing-surfaces in this joint is comparatively small, so that the specific pressure may be very high: it varies throughout the stroke, but its prevailing maximum value will be around 1000 lbs. per sq. in. in stationary engines, and will rise to 4000 lbs. per sq. in. in heavy-pulling locomotives, where 6000 is often reached

when the heaviest traction is being exerted, at low speed. The marine engine, with two wrist-pin journals, need not have much more intense pressures than are usual in stationary practice.

*Holding the Pin.*—Two examples of cylindrical pin-fits have already been noted, in Figs. 282 and 284. Fig. 290 shows another with a very neat and simple arrangement for clamping it; here the pin must be made an easy driving fit, so that screwing up the clamping bolts will not sensibly distort the cross-head. With the pin fixed in a forked rod, as in Fig. 296, a cylindrical shrink-fit :

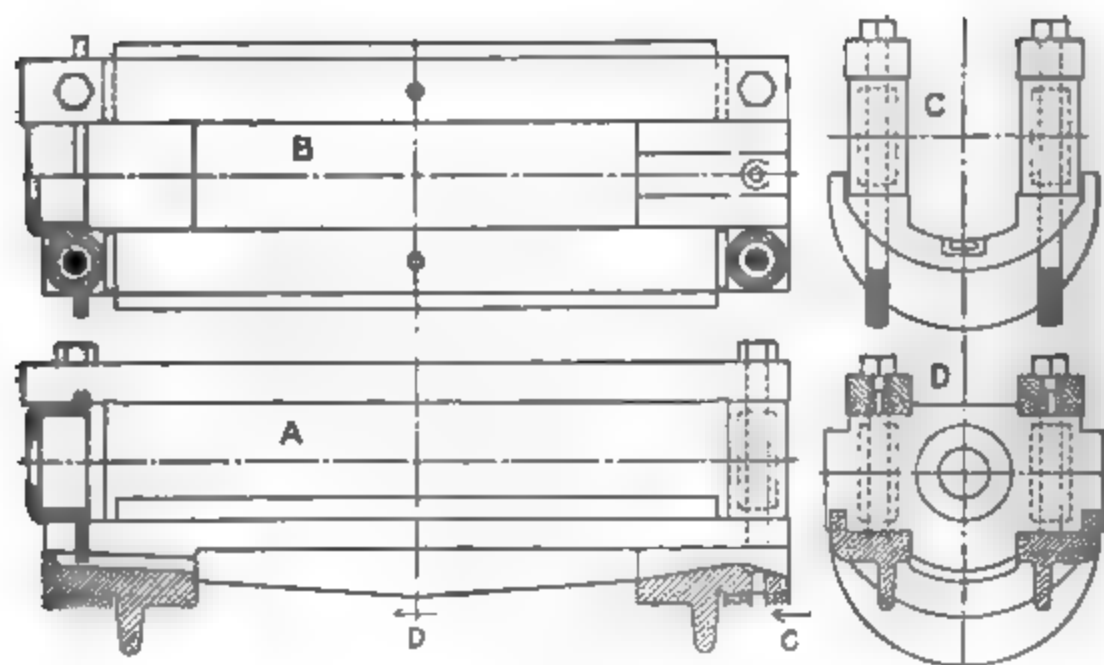


FIG. 298.—Guide-bars to fit Figs. 237 and 282. Scale 1 to 14.

generally made; although when this form of rod is used in stationary engines (in English practice), the pin has sometimes a taper joint and a retaining nut.

Our examples convey the correct impression that this last method is the one most used for holding the pin in the cross-head. We distinguish two cases: most usually the two taper-fits are part of one cone, so that the holes can be reamed with a plain taper reamer; the only example of the other case is seen in Fig. 291. This last shows also a way of holding the pin in place that is different from the usual nut on the end of the pin; and a smaller variation from the latter method is seen in Fig. 285. Sometimes though by no means always, a shoulder on the pin limits its wedge



action; but there is far less reason for it here than in a piston-rod joint, where the major force-action is along the axis of the cone. Very often there is a little key on one side of the head, to keep the pin from turning.

(c) GUIDE-BARS.—In the great majority of stationary engines, the guides are wholly or in great part formed right upon the frame-casting. On locomotives the bars are usually of forged steel, supported as shown in Figs. 231 and 246. Several examples of

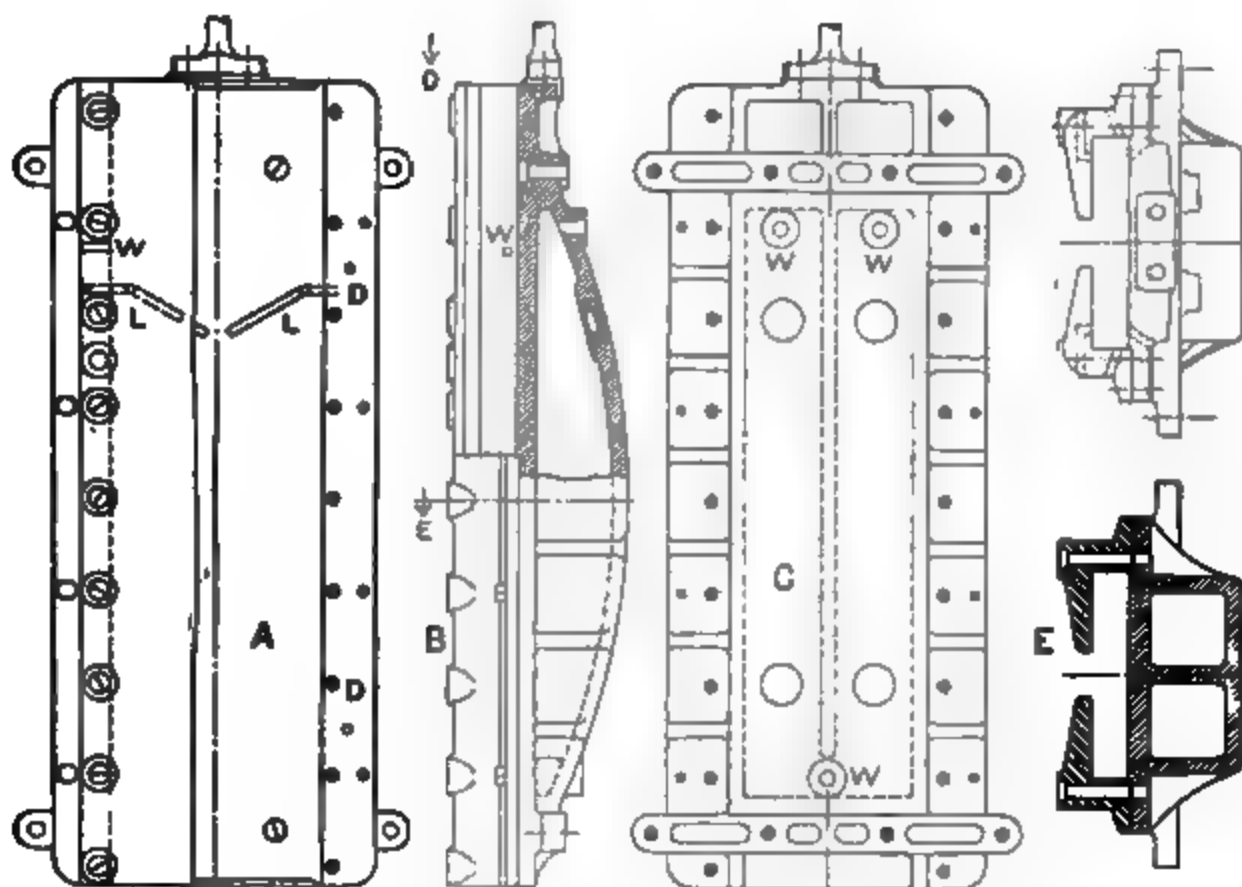
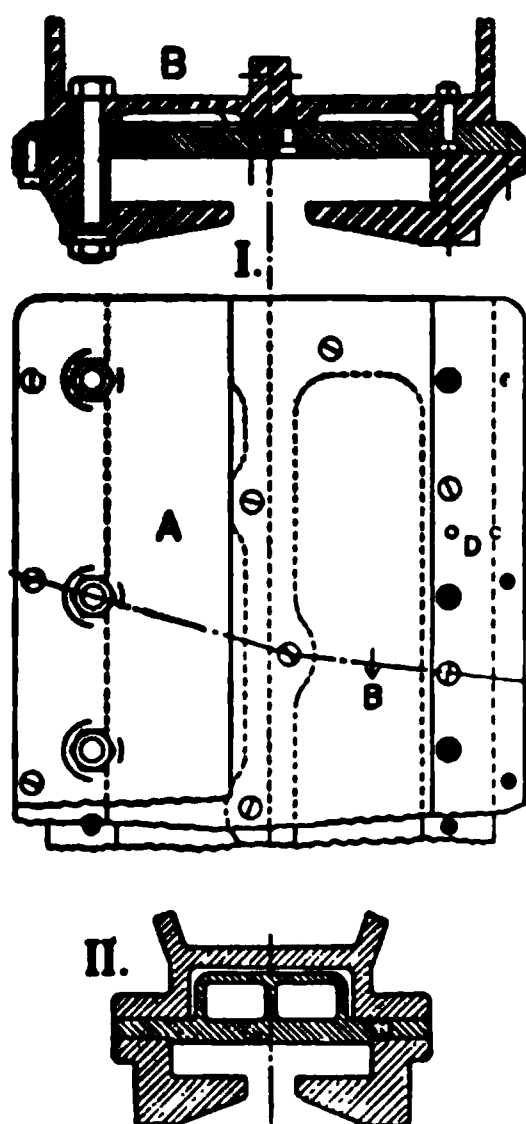


FIG. 299.—Guide for Warship Engine, with frame of steel columns; cross-head in Fig. 295. Scale 1 to 24.

separate guides of cast metal, and hence of more complicated form, will now be illustrated. The first, in Fig. 298, is an enlarged detail from Fig. 237, for the cross-head in Fig. 282. The bottom bars are joined by cross-pieces at the ends, so as to form a single casting, which rests on bored seatings in the bed. Columns or space-blocks support the top bars (likewise of cast iron), those nearer the cylinder being joined by a light cross-web which serves as an oil-guard.

Guides for marine engines with slipper cross-heads are shown Figs. 299 and 300. The first is very much like those on the engine in Fig. 235, except that in this both ends are supported by cross bars bolted to the columns, instead of the top end being fastened to a bracket formed upon the cylinder-head; a short strut at the top helps to resist any tendency to vertical movement, which might be induced by the frictional drag of the cross-head. The main guide, piece 1, is of cast iron; this metal gives a rather better wearing-surface than would steel, and it is far easier to make a good hollow casting with iron: but the front or backward-running guides are of steel for greater strength. When lifted into place these bars are at first secured by the light tap-bolts near the out-



edge, then strongly fastened with the heavier through-bolts. Under the bars are several liners of sheet-metal, located by the dowel-pins at D, D. The water system is clear, with the connection marked W in view C: besides the circulation in the water-back, a direct spray can be thrown upon the rubbing-surface through the holes in the side-bars, as at W in view A. In this view the oil holes and grooves are shown at L.

In Fig. 300 the main Y-frame is cast steel, made in two parts with bolted flanges. The guide-plate of cast iron is fastened in place with a number of sun-head bolts: and after the cast-steel front guides have been screwed fast, heavy through-bolts are inserted to hold the whole structure rigidly together. Water spaces are formed between guide-plate and frame, with due provision for support and circulation; and the sketch at II.

given as illustrating another way of forming this water-back.

In Fig. 293, view C, is shown a section of the separate guide-bar which is bolted to each of the four cast columns that make up the

FIG. 300. — Marine Guides for Cast Columns: I. From Fig. 232; II From Fig. 245. Scale of both, 1 to 30.

frame under one cylinder and this principle, of forming the guide surface upon a light separate piece, is applied almost without exception in marine practice.

Compare with these marine arrangements the device used in Fig. 289, where the guide, relatively very heavy, is divided in the plane of the engine mechanism and the halves are strongly bolted together, of course the supports at the cylinder-head and at the guide-yoke help to keep these halves rigidly in position so that their surfaces will form a true plane.

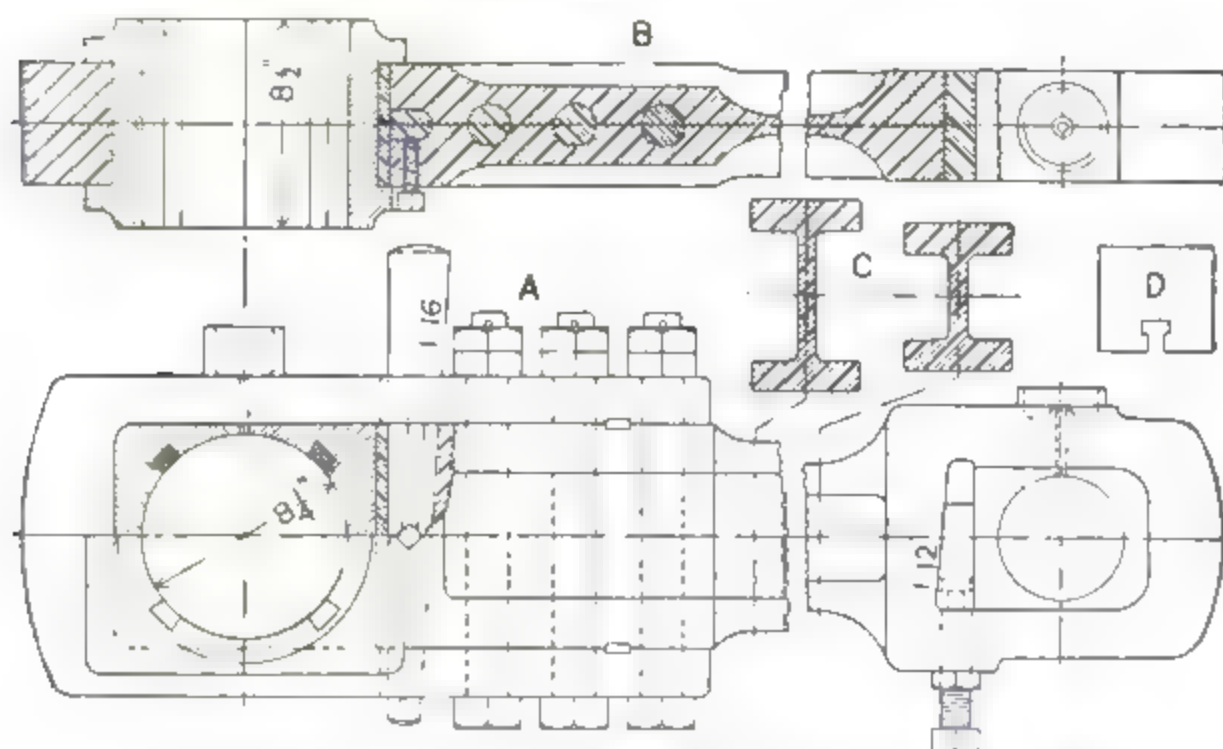


FIG. 301. Locomotive Connecting rod cross head in Fig. 288.  
Scale 1/12.

(p) **CONNECTING-RODS.** Without running over Figs. 301 to 311 in sequence and noting individual particulars, we will consider the important details collectively, taking up first the crank or foot of the rod. This part describes transmitting alternately a force alternately tensile and compressive and also resisting the forces due to gravity and inertia. For saw-milling engine of the Corliss type the shank is round, usually tapered toward the ends and tapering slightly toward the heads as in Fig. 301, 309, or 313. With higher speeds, the rod may be either a long one, tapering toward the crank end and flatted at the ends as in Fig. 302, or a

rectangular cross-section as the diameter increases—see Figs. 219, 220, 221, and 229. Stationary high-speed engines, and great many locomotives, have the rectangular section, increase in depth toward the crank-end, as seen in Figs. 303, 305, 203, 223. For high-speed locomotives the sides of this rectangular are milled out, making the I-beam cross-section shown in Figs. 301, 302, 306 I., and 231. The marine engine runs at a low rotative speed; but the rods are relatively short, the bending action is less severe, and the body is made round, with a slight taper.

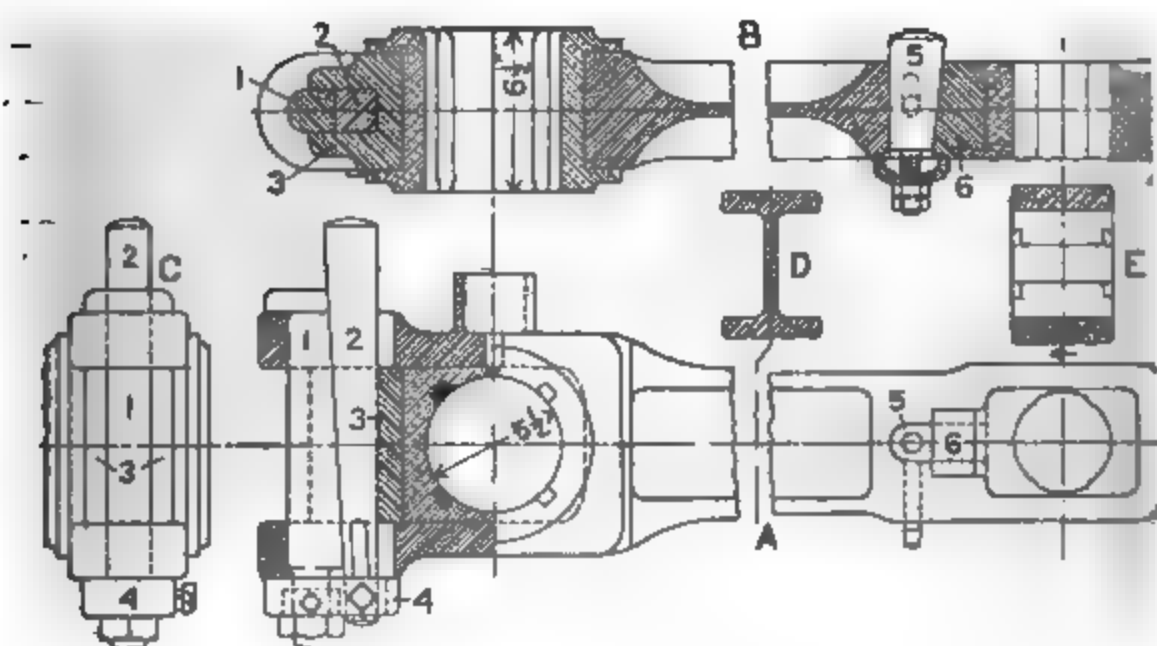


FIG. 302.—Rod for Express Locomotive, cross-head in Fig. 289.  
Scale 1 to 12.

In stationary engines the ratio of rod-length to crank-radius usually 6 to 1 or somewhere near that value; in locomotives where the front driving-axle is not the main axle (as in Fig. 231), the ratio ranges from 7 to 10; in marine engines it is usually about 3 to 1 has been used in extreme cases, but this greatly distorts the motion of the piston from symmetrical harmonic motion.

(g) CONNECTING-ROD ENDS.—At each end of the rod there is an adjustable bearing for one of the pins. The “boxes” or “brasses” which form the bearing proper must be enclosed in a frame or casing of suitable form and strength, and provision must be made for setting them to a proper fit upon the pin and for taking

up wear. In the construction of the rod-end which encloses the bearing, four types may be differentiated among the examples given. The solid end is seen at the wrist-pin in Figs. 301, 302, and

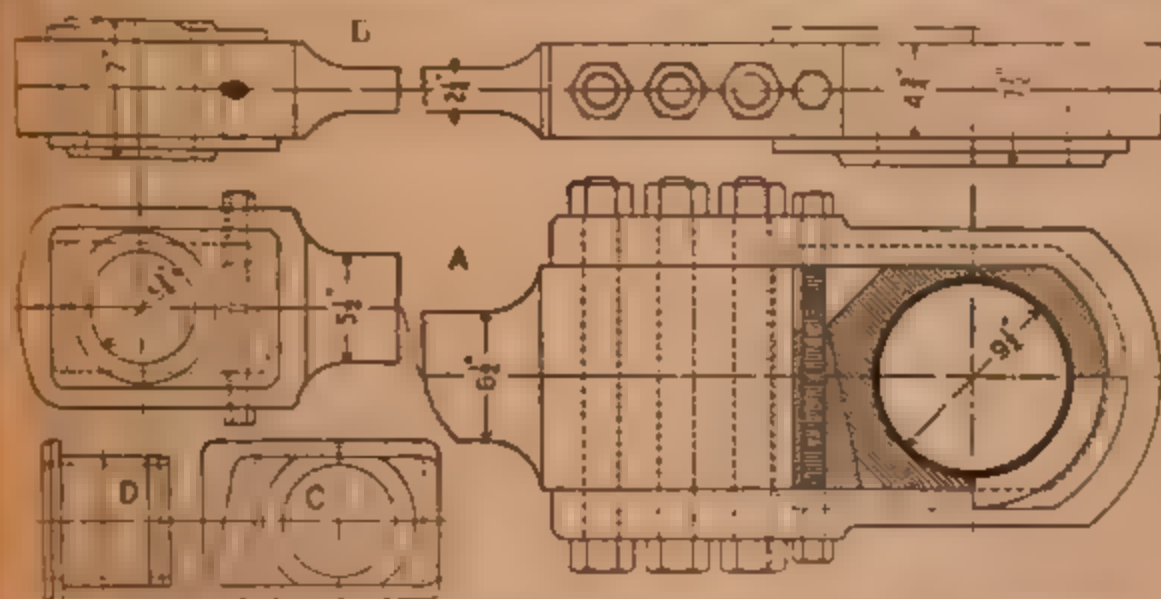


FIG. 303. - Rod for 22'' and 42'' by 27'' Engine, 175 R. P. M., pistons in Fig. 279 I. Scale 1 to 15.

303, and at both ends in Fig. 304. An example of the *jaw* end is given in Fig. 302, and slightly different arrangements are outlined in Figs. 263, 391, and 394; it is especially a German type. The very common *strap* end is illustrated in Figs. 301 and 303, with

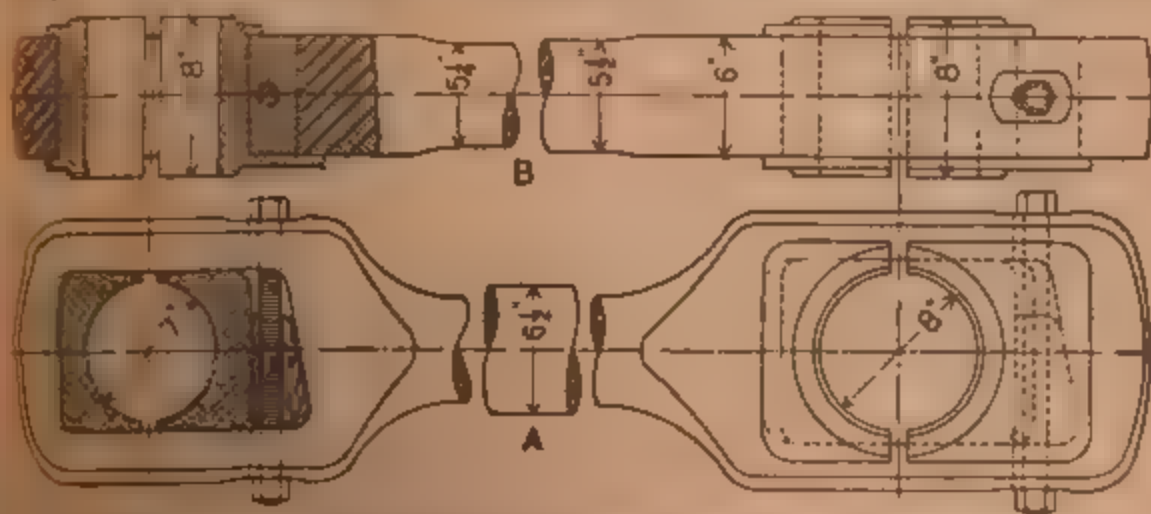


FIG. 304. Rod for 26'' by 48'' Corliss Engine, Figs. 218, 241, 277. Scale 1 to 15.

modified forms in Fig. 306. The *marine* type, Figs. 305, 308, etc., is used practically without exception on marine engines and very largely in stationary practice.

In the old-fashioned key-and-gib strap-end (see Fig. 451 I) the two functions of holding and adjusting the bearing were combined in one arrangement; but in most modern designs there is a strong, rigid enclosure, within which the adjustment is made. An exception to this statement appears, at first sight, in Fig. 302, but vanishes when we note that the gib-bolt 1 does not move, and that the key 2 and the face-block 3 together form the adjusting device. A real exception is furnished by the marine end; but there the conditions are different, in that the take-up is along the axis of the main holding-bolts; and further, since the boxes are usually separated by liners or shims, and the whole bearing then bolted hard together, the adjustment that can be made by changing the liners is very different from that made by a key or wedge.

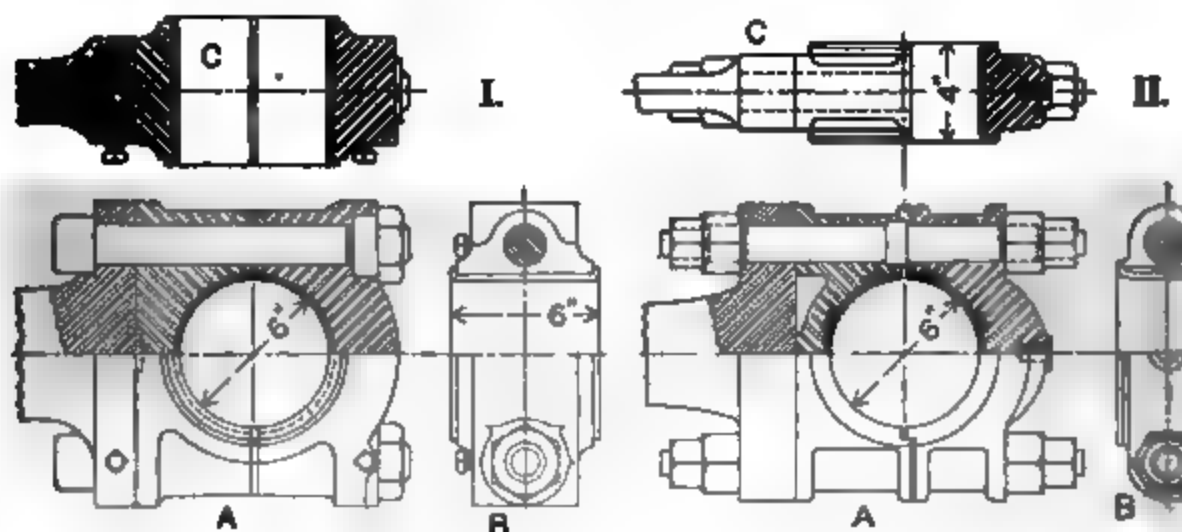


FIG. 305.—Marine Rod ends for Stationary Engines: I. For 15" by 1" engine, Figs. 201, 251; II. For 16" by 16" engine, Figs. 206, 238. See 1 to 12

*Types of Construction.*—Looking at the enclosed ends a little more in detail, we see in Fig. 303 a simple bolted strap, which held against tension along the rod wholly by the shearing strength of the bolts (plus whatever friction is developed by their grip). In Figs. 301 and 306 II. this shear is largely taken by keys. The strap, besides resisting direct tension along the rod, must also have a very considerable amount of strength and stiffness against transverse forces, central at the crank-pin; which explains why it is made so much heavier than simple tension would seem to require. In this respect the jaw arrangement of Fig. 302 is at



decided advantage, because the sides of the enclosure are held together clear up to the box. In Fig. 306 II. the bolts are supplemented, as to this duty of holding the strap together, by the dovetailed keys and by the light binding strap; and then the

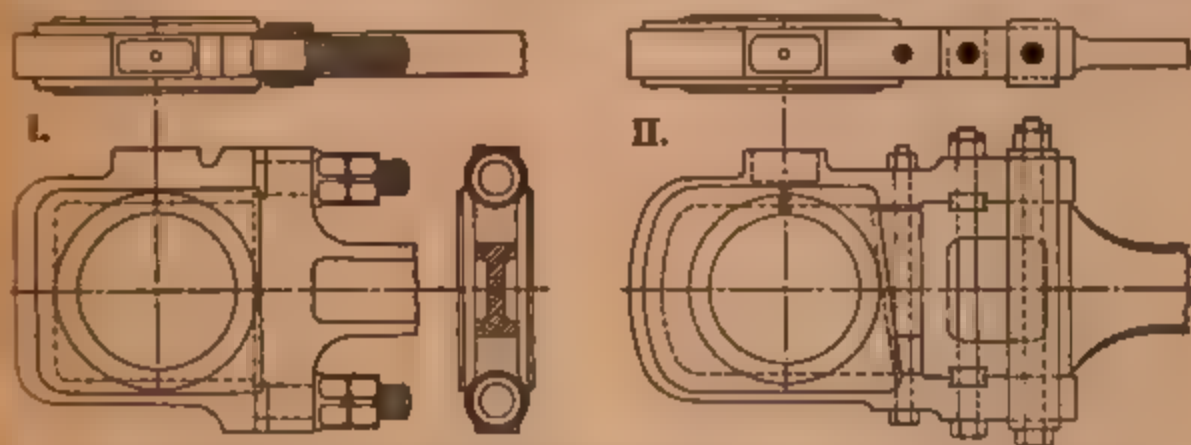


FIG. 306. —Special Rod-ends, French locomotives with inside cylinders.  
Scale 1 to 16.

comparatively small bolt-holes do not remove too large a fraction of the metal in a cross-section of the rod and strap. The arrangement in Fig. 306 I. is really a combination of the fundamental ideas of the marine and of the strap end. In these last two designs the bearing is so short, relative to its diameter, because the pins of these inside cranks must be kept at the full diameter of the axle;



FIG. 307.—Connecting-rod of Westinghouse Single-acting Engine,  
Figs. 207, 208.

then sufficient bearing-surface can be got with a short pin, and besides there is not room for a long one within the limits of the locomotive.

On the coupling-rods of a locomotive, solid, non-adjustable bearings are used; but always the connecting-rod of any engine is provided with means for taking up wear.

(r) A SPECIAL DESIGN, for a single-acting engine, is shown in Fig. 307; it is peculiar in that one long strap goes around the whole rod, binding the parts together. All the heavy force transmitted by the rod is compressive, so that the strap is never subjected to a tensile stress of any magnitude. It is strongly bolted to the

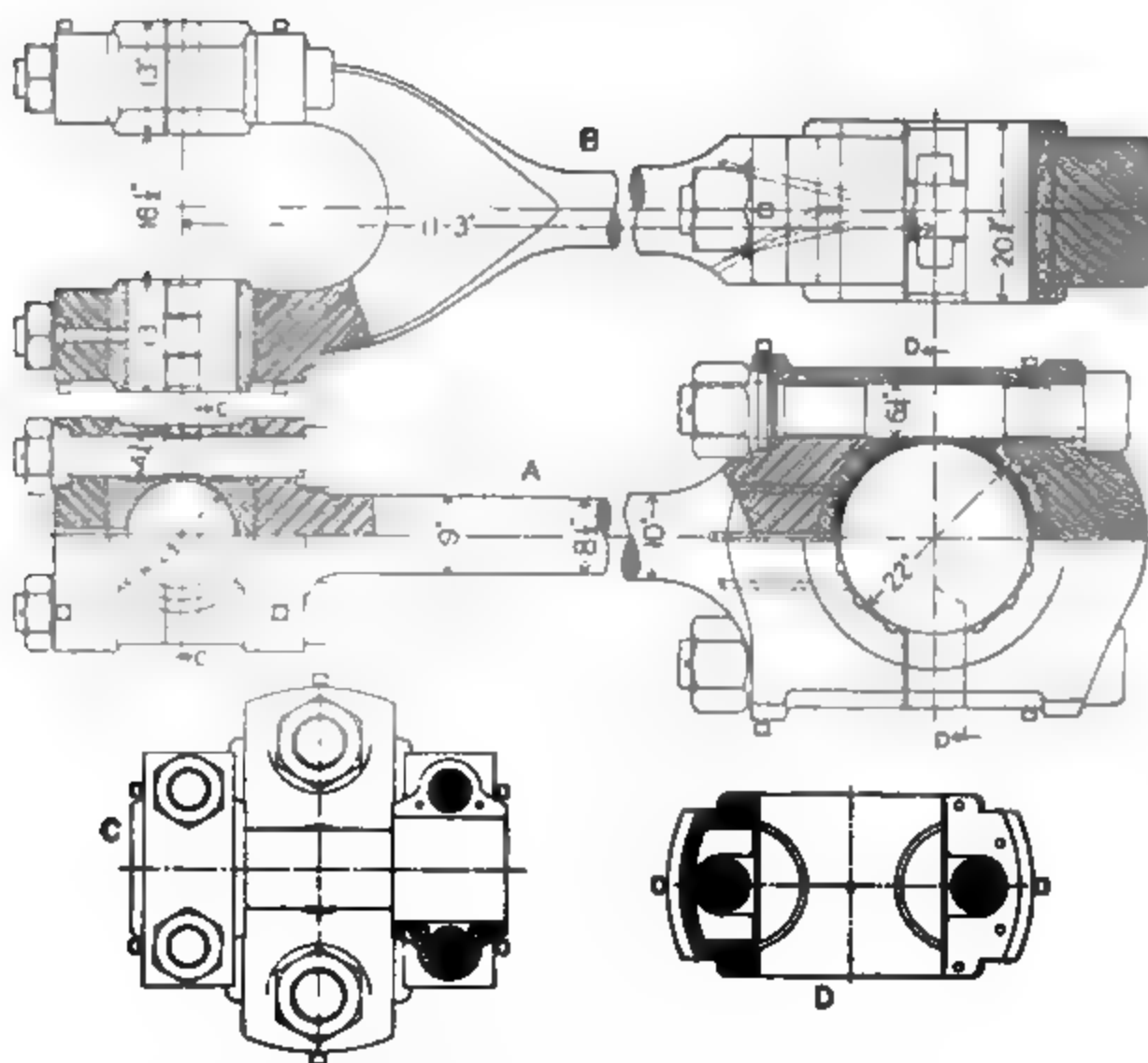


FIG. 308.—Rod from Fig. 232. Scale 1 to 32.

outer box at the crank-end, and all the adjustment is made by one wedge; and this wedge has flanges which keep the rod-body from getting out of place sidewise. The hole marked C is used only when the rod is being put together and handled; a pin stuck in here holds together all the parts above (to the right of) this point. In this rod, as in Fig. 304, the adjustment of the bearings is made entirely by "feel," as there is no clamping of the boxes edge against edge.



(s) MARINE CONNECTING-RODS.—Various forms of the marine end are shown by Figs. 308 to 311, one typical arrangement at the crank end in 308 and at 310, the other at the cross-head end of 308 and at 309. In the first, the rod-end and cap form a circular eye or enclosure, within which is held the light cylindrical shell that makes the bearing: this type of end is usually forged as a solid head, the opening for the bearing is made by drilling holes in a circle so as to set free a central core, and finally the cap-piece is cut off, in which operation enough metal is removed to make room for the distance-piece. As to the profile of a lengthwise section of the bearing, the two examples given show quite different shapes. These rods, when for high-speed engines, are always very closely designed for strength, so as to minimize their weight; and Fig. 310 shows especially close trimming.

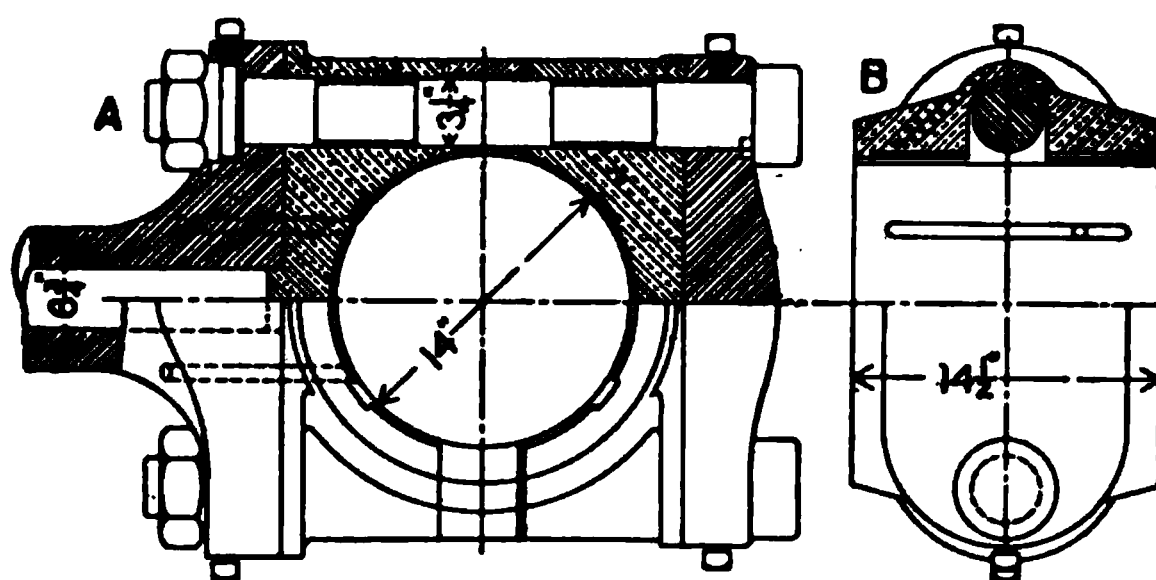


FIG. 309.—Rod with T-end and Square Boxes, cross-head in Fig. 295.  
Scale 1 to 18.

The second type of head has separate rectangular boxes, held between a T-end forged upon the rod and a flat cap also of forged steel. These boxes are of brass or bronze, lined with white metal at the crank-pin, but with the harder surface of the copper-alloy at the wrist-pin, where the specific pressure is so much higher. Most of the adaptations of this rod-end to stationary engines are like Fig. 305 I.; 305 II. is peculiar in the general form of its bolts and in the detail of the lock-nut arrangement. The boxes are here of cast steel in both designs, although cast iron is often used; and the outer box is made strong enough to serve as cap also.

In Figs. 308 to 311, note how the bolts are kept at the full

diameter, over the screw-threads, wherever they act as do pins to keep the parts in alignment; but are elsewhere turned down to the effective diameter inside the screw. In Figs. 308 and 309 the bolt is kept from turning by a key-pin under the head and is held in the cap by a set-screw; and the same nut-locked device is used in all these designs. The distance-pieces in 308 are of cast iron; one is shown in section at D, held in position by four dowels. That in Fig. 310 is a light brass casting.

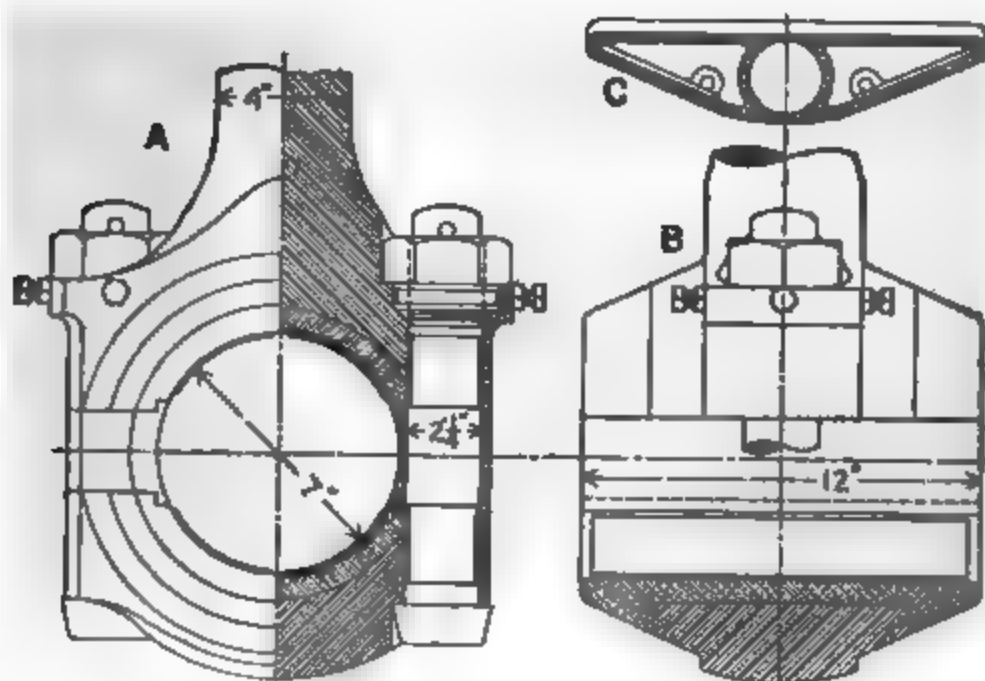


FIG. 310.—Rod with Round Eye, British torpedo-boat destroyer, cylinder 19½", 30", 34"–34" by 18". Scale 1 to 9.

nearly every case, thin liners are used for adjustment; with these it would be necessary to file or plane down the distance block as the bearing wore loose.

Fig. 311 shows an adaptation of the idea of Fig. 306 I. to a large marine engine. By the combination of the cap with bolts into one piece, the minimum of weight seems to be secured but this semicircular strap is a rather awkward piece to machine.

(f) ROD-BEARINGS AND THEIR ADJUSTMENT.—For stationary engines, the crank-pin boxes are usually of cast iron or cast steel with babbitt lining. Only in the locomotive are they regularly made of brass or bronze, which bears directly upon the pin. In the wrist-pin, brass bearings are generally used. Brass boxes with babbitt belong especially to marine practice, as already remarked.

The lining of white metal is almost always cast in place, and held by dovetail grooves. An exception is seen in Fig. 305 I., where there is a loose thin shell of babbitt, set into the boxes and kept from turning only by the liners at the joints, which project inward. The best white metal is composed mostly of tin, with some copper and antimony: in cheaper grades, more or less lead is used. In large bearings the lining is well hammered after

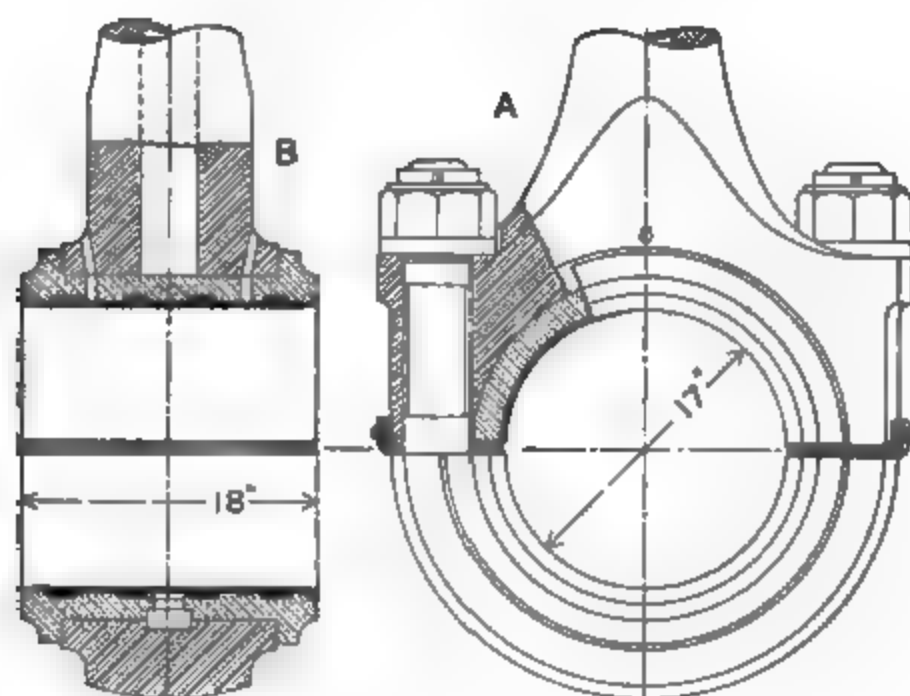


FIG. 311.—Marine Rod with Semicircular Strap, Schneider & Co., France.  
Scale 1 to 18.

casting, to harden it and force it closely into the grooves, and is then bored out to the exact diameter for the journal. This statement applies to new work; a repair job means usually simply casting the lining around the journal, without any other finish.

The boxes are held against sidewise displacement by flanges, which extend all around the enclosure when the latter can be opened to insert the boxes, as at the crank end in Figs. 301, 302, and 303. With a solid end, the flanges on one side cannot be complete, but must be made as shown at C and D in Fig. 303, where the inner box, with the wedge, is slipped into place last and then held by the wedge-bolts. In Fig. 301 the wrist-pin boxes are held in simply by the cross-head; in Fig. 302 the block 6 is first put in, then the box next to it, lastly the outer box.

*Adjustment-devices.*—The use of a key to adjust the bearing, in Fig. 301, is quite common on locomotives. It has the disadvantage that there is only a comparatively small bearing surface of wedge on box, and the metal is likely to be crushed in course of time. In the example shown, this difficulty is overcome by putting a steel plate between the key and the bronze box; in Fig. 302 the key is unusually thick, and bears upon the steel face of block 3. Generally, however, a broader wedge is preferred, as in Figs. 303 and 304 especially. Various ways of holding and adjusting the wedge are shown, all of them securing it against motion in both directions. Note that in the solid ends the slanting bearing surface is formed upon the rod; while an example of the opposite arrangement is seen at the crank-pin in Fig. 303.

*Lubrication.*—The provision for lubrication consists of oil-holes and grooves; the former come either from the outside of the bearing or from within the pins; the grooves are cut in the lining of the bearing, and are generally short segments of a helix, so as to lead the oil over the whole surface. In most of the drawings the lubrication details are omitted, in favor of a clearer view of the details of construction. On marine engines, oil-pipes are fastened along the rod, entering the bearing through a drilled hole: at the top, near the wrist-pin, the pipe carries a small box with a wiper which takes oil from a fixed pipe. A part of this arrangement can be distinguished on Fig. 235.

#### § 45. The Rotating Parts of the Engine.

(a) SHAFTS WITH (OVERHANGING CRANK.—The usual type of shaft for stationary side-crank engines is well represented by Fig. 312 I. Shaft and crank-pin are of mild steel, the disk is of cast iron with a fan-shaped counterweight formed upon it. They are put together with either forced or shrunk fits, the holes in the disk being bored about one in one thousand smaller than the pieces which are to go into them. The crank-pin is riveted over for greater security, and has a detachable cap, to receive the solid end of the connecting-rod shown in Fig. 304. To carry the weight of the fly-wheel and generator, the shaft is enlarged between the bearing

but its diameter can be reduced where it enters the crank-hub without sacrifice of needed strength, although the proportion of reduction is greater in this case than where the conditions of working are more severe.

Fig 312 II shows the shaft for a large duplex horizontal-vertical engine. The straight shaft is a hollow forging of high-grade steel, oil-tempered; the disks or webs are massive steel

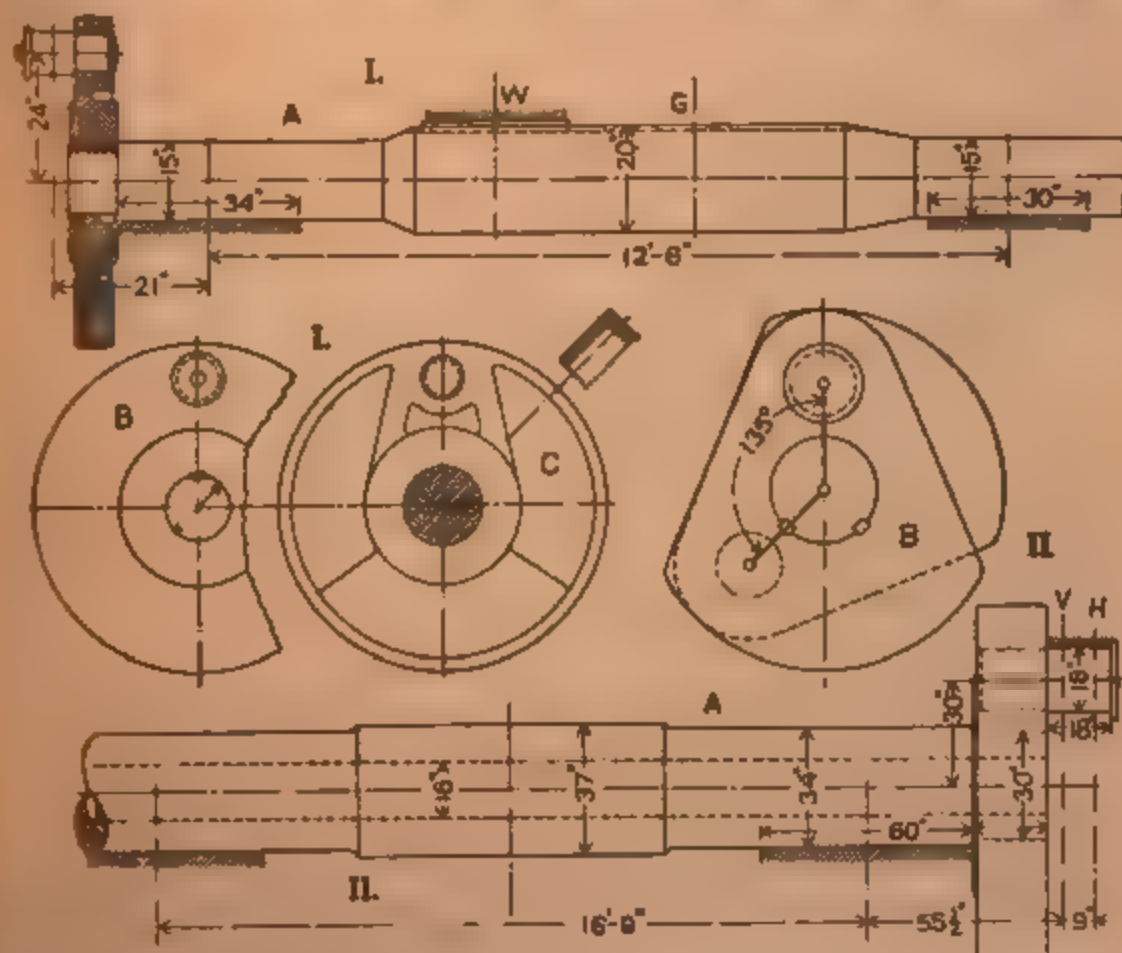


FIG 312—Shafts with Built-up End-cranks. I For 26'' by 48'' Corliss engine, Figs 213, 241, scale 1 to 48; II For engine in Fig 219, scale 1 to 72.

castings. Referring to Fig. 219, we see that the two connecting-rods act upon the one pin as here indicated by the letters H and V.

A rather special example is given in Fig. 313; the crank is made, not of ordinary cast iron, but of a mixture called *steelled cast iron* or *semi-steel*—steel being added to the iron in the melting, so as greatly to increase its strength and toughness. The hole in the middle of the pin is cored out chiefly to help in getting a sound casting: when plugged it forms part of the oil-conduit. Note the



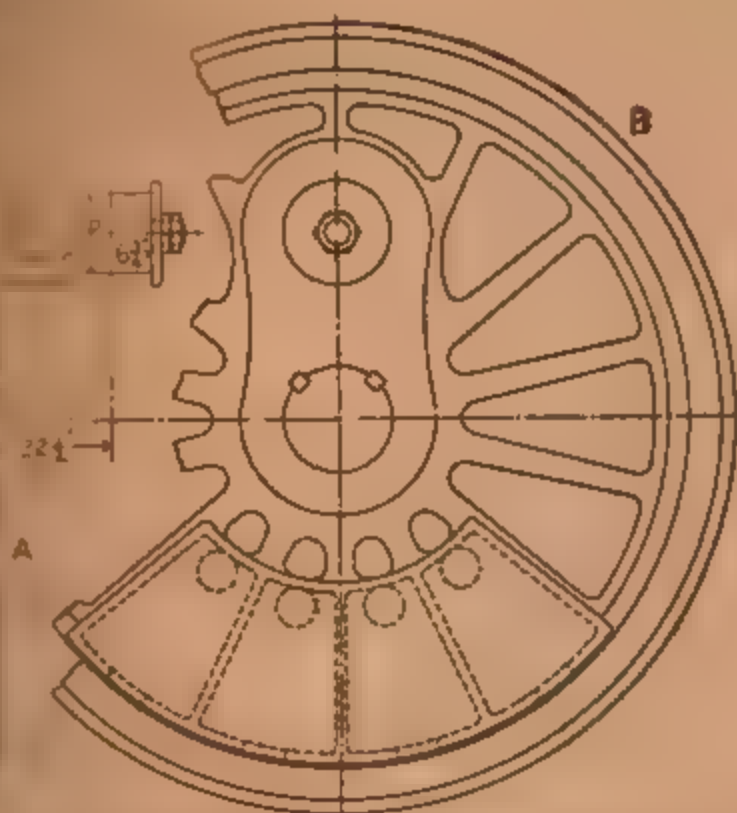


Fig. 314—Shaft for a "Consolidation" (Four-coupled) Locomotive, in Fig. 256 and about 40,000 lbs. on each driving-wheel.

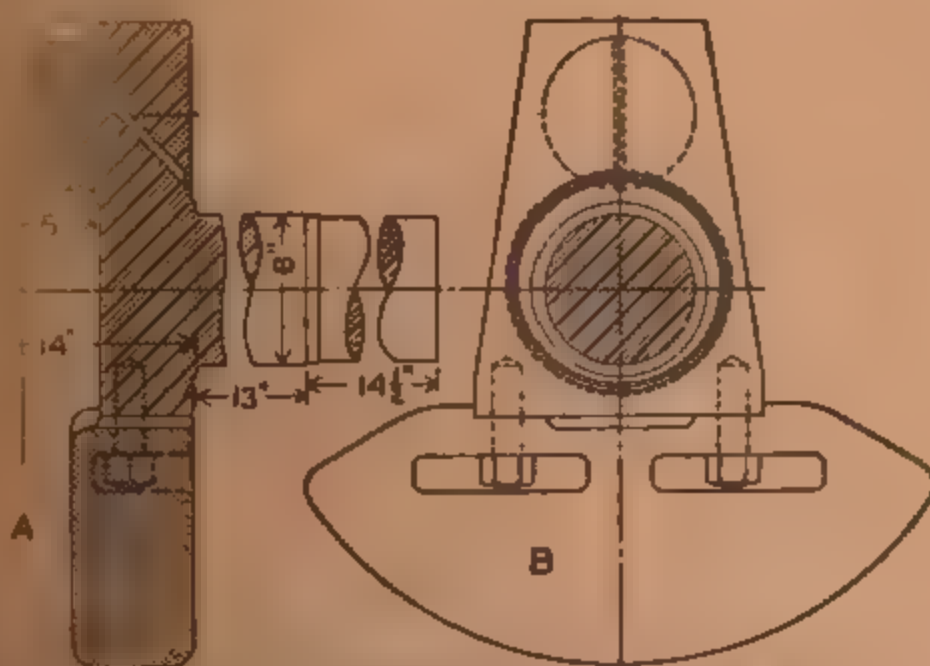


Fig. 315—Shaft for 15" by 14" High-speed Engine, Figs. 201, 237. Scale 1 to 12.



## CONSTRUCTION OF THE ENGINE. [Ch. VIII]

- amount of metal beyond the shaft; and it is required amount of counterbalance by the weight of the connecting-rod is much more accessible than in the case of regular crank-disks. Of course, there is a danger that the weight may get loose, which must be attended to at not too long intervals. When a counterbalance is provided than can conveniently be provided the crank-disk may be made hollow and filled with lead, as in the case of this type are of steel, the crank being cut out in the machine-shop; but counterweights have been used for small shafts, say under 2 inches, with satisfactory results.

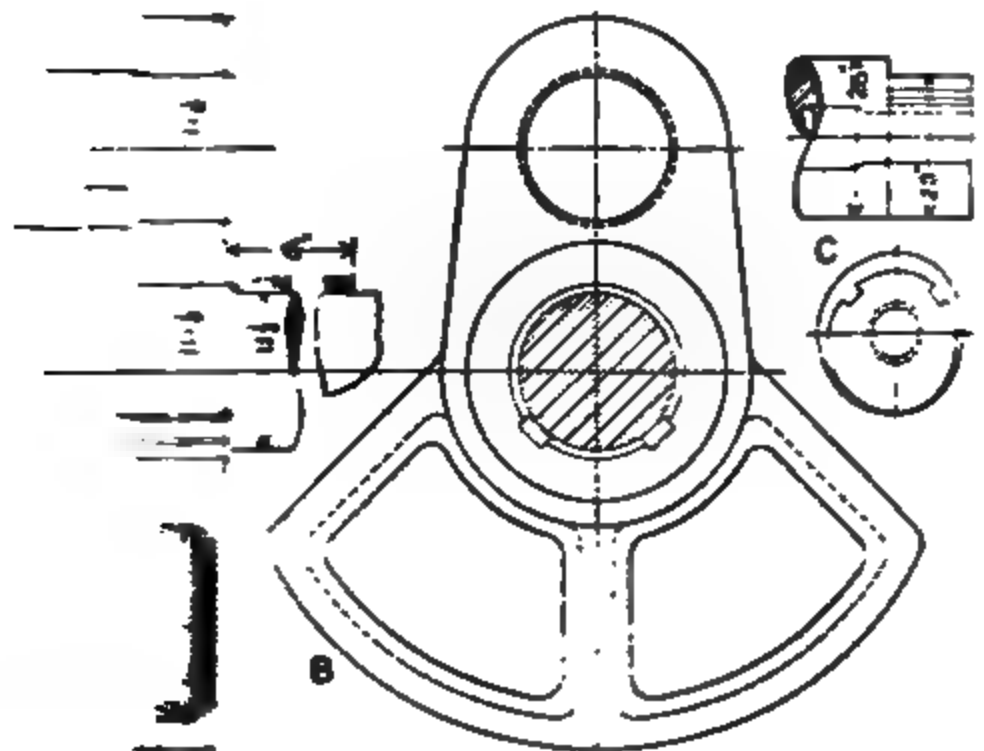


Fig. 217. Crank-disk for 24" and 42" by 27" Vertical Engine. Figs. 217, 218, 219. Scale 1 to 16.

- In a counter-crank compound engine is shown the feature of the arrangement of the ends of the connecting-rod—this being the case in Fig. 218. For multiple-crank stationary engines the counterweights are arranged in the same manner as in the case of the crank-disk.



With a large shaft and crank-pin there is likely to be very little metal left between the holes in the crank-arm. One way of overcoming this difficulty is shown at C in Fig. 316,\* where the reduced end is made eccentric to the body of the shaft, being moved away from the pin.

(c) SHAFTS FOR MARINE ENGINES.—For very large marine engines, the built-up type of shaft shown in Fig. 317 is generally used. All the parts are of high-grade steel—nickel-steel in this particular case—and they are put together with shrunk fits. Quite often the pins are keyed as well as the shaft-ends, but there

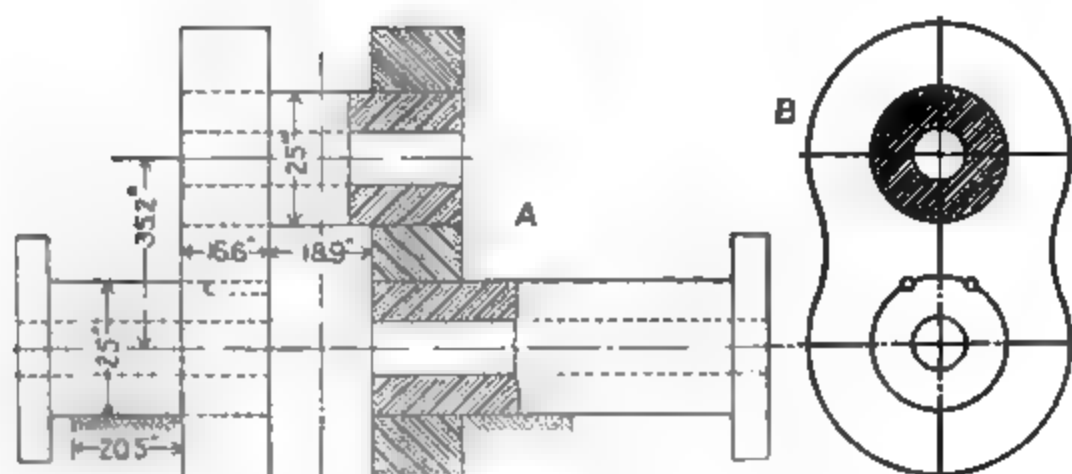


FIG. 317.—Section of Shaft for Large Passenger Steamer, pistons in Fig. 279 II. Scale 1 to 56.

is really no need for this, because the pin has no tendency to turn in its hole unless the bearings fail radically in their duty of properly supporting the shaft. Removing the core saves weight with only a minute decrease from the strength of a solid shaft of the same diameter; when the piece is large enough to be forged hollow, upon a mandrel, it decreases very much the thickness of metal acted upon by the hammer or the forging-press; and it greatly facilitates the heat-treatment of annealing and tempering upon which the quality of this class of forging so largely depends.

In engines of medium size, for war-ships especially, the cranks are solid-forged, and very often there are two in one section, as in Fig. 318 I. An important advantage of the solid crank is its

\* Westinghouse Machine Co., see *Power*, May 1902.

centralizes an equal amount of weight on each side, and is therefore easier to get a round shaft. The construction here given is that the end of the crank is not so far from the center as when it is between cranks, and there is a larger possibility that the crank will be broken, guarded against by inspection. A greater counterforce is needed with iron, the weights are more numerous. Fig. 314. Usually, shafts are forged as a solid block, but quite frequently steel castings of the size in Fig. 315 and

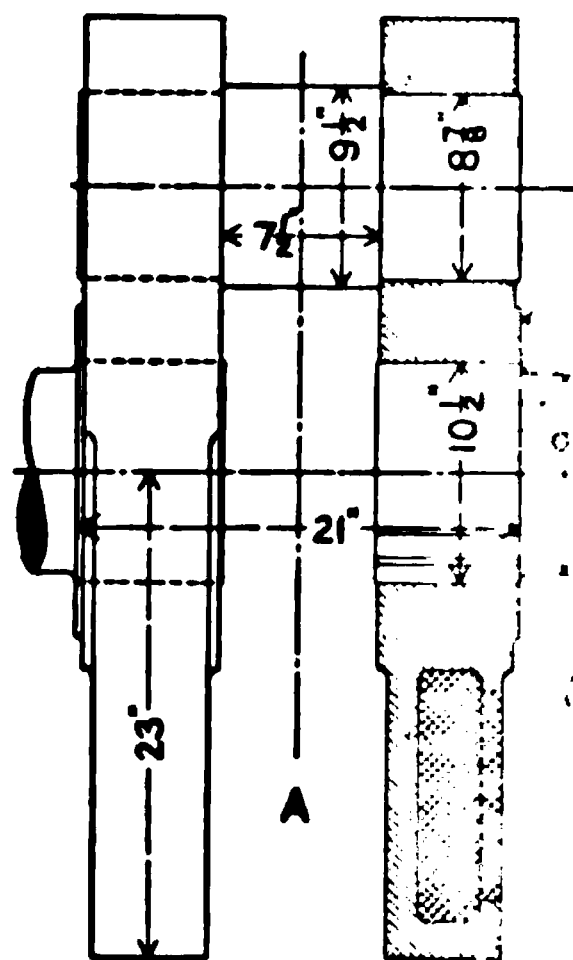


FIG. 316.—One Crank for a shaft.

A part of a shaft for a torpedo-boat engine is shown in Fig. 316. A noticeable feature of the shaft-sections which are in marked contrast to those of engines, this built-up type is the solid-forged type and

the built-up type. With the cranks in two closely-spaced positions, it is not possible to get a joint that a joint could not be made in any case. the shaft is made of steel and joined by flange-couplings. See Figs. 232 to 235; and details of the shaft in Fig. 317. Sometimes cylindrical or

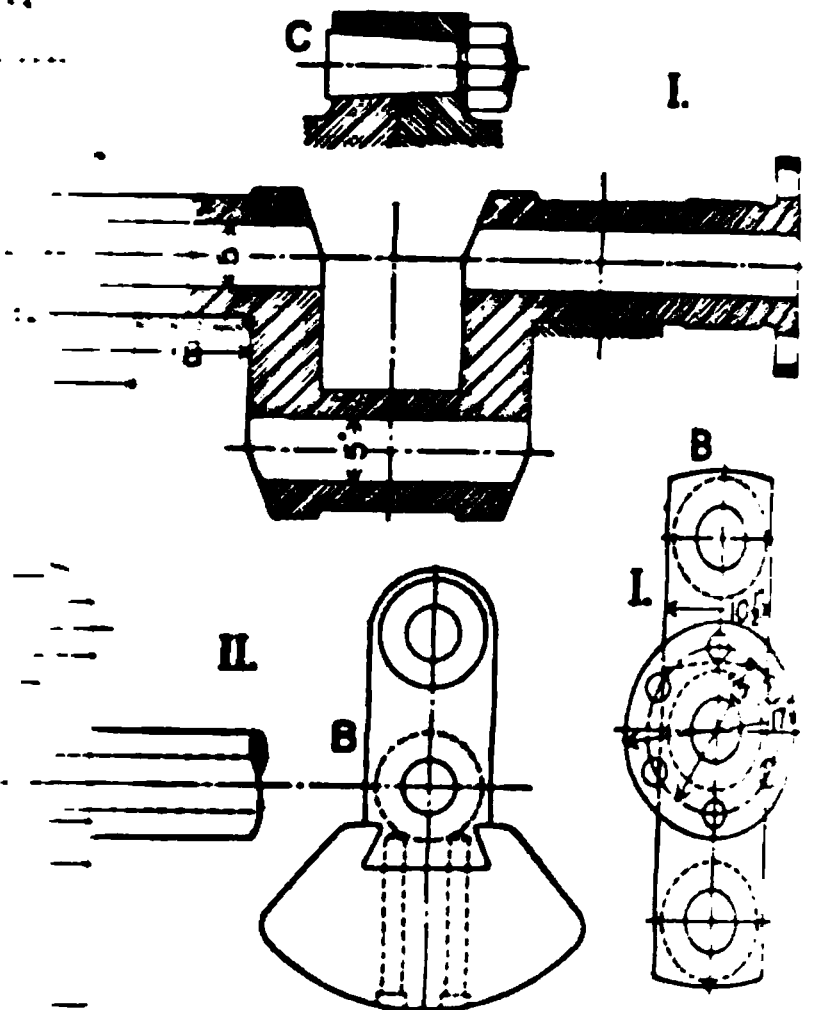


FIG. 317.—Details of a shaft. I. For U. S. Cruiser, "Denver" 10" by 30", scale 1 to 30; II. For U. S. Cruiser, "Denver" 16" by 17", scale 1 to 37.5; III. For U. S. Cruiser, "Denver" 10" by 30", scale 1 to 30.

the conical bolt here illustrated is especially is it easier to take out on a large size is a tendency to enlarge the hole. It is not difficult to provide enough strength against this action.

A torpedo-boat shaft, with counterweights, these weights must be very light. The highest speed of this engine is about 1000 revolutions per minute with the same stroke. In small engines of service, the shafts are made

piece, with three or four cranks, and sometimes even eccentrics forged upon them. This is very largely a matter of absolute size and weight, facility in handling, both in the workshop and in the ship, having a great influence in determining what shall be put into the unit.

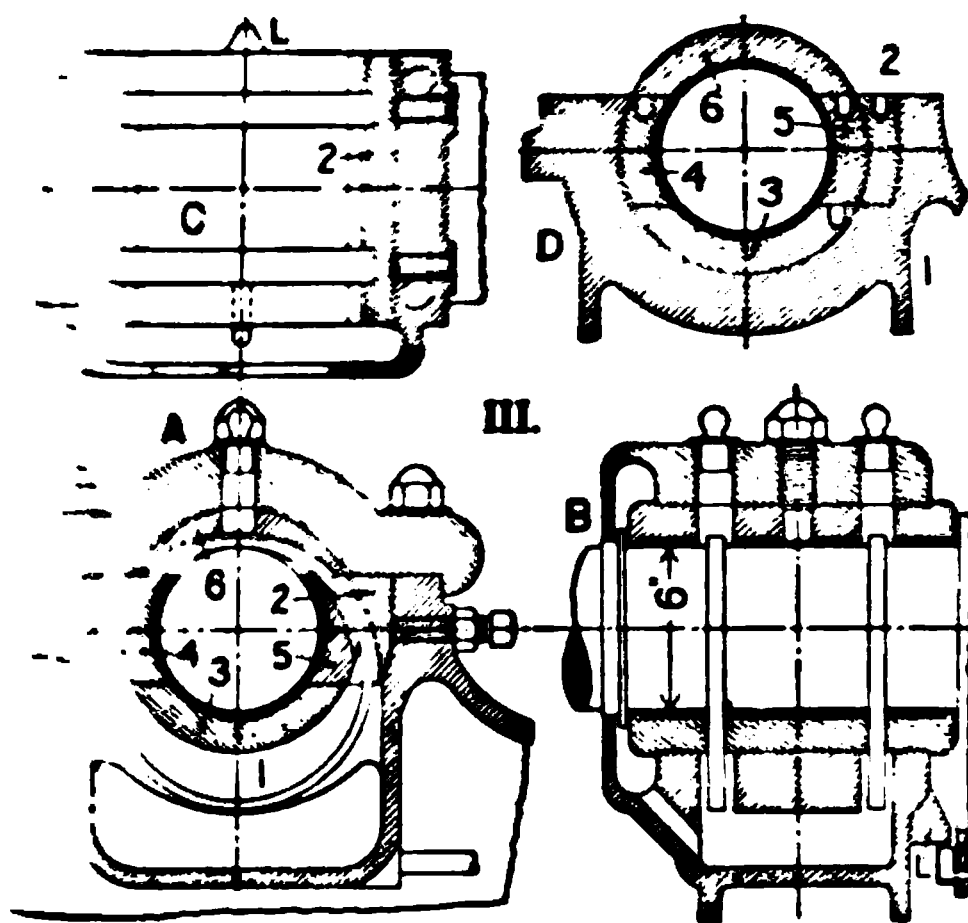
It is sketched a flange-joint with a key, intended to relieve the bolts of shearing stress. As these bolts can easily be made of ample strength, there is little need for this extra piece, and it is seldom used.

**STRENGTH OF THE SHAFT.**—With the engine-forces known, it is possible to determine quite closely the stresses developed in the shaft, graphical methods being particularly advantageous. Since there is a combination of bending and torsion, so that the problem, though capable of rational solution, is likely to be very complicated, especially is this true of a shaft with a number of cranks. The extreme case is the engine to which Fig. 317 belongs. In this engine there are two three-crank engines coupled fore and aft and driving one propeller, or six cranks in all: close analysis would give the required diameters for shaft and crank-pins, going aft. Actually, the forward engine has a shaft diameter of 21 ins. against 25 ins. in the after engine. All the crank-pins are of the same size, for the purpose of having the connecting-rods alike; but the axial hole is larger in those which have the smaller force to transmit.

We shall not take up the problem of shaft design at this point, but mention more than the other matters of design for strength. But as an indication of the importance of being able to solve problems relating to the strength and stiffness of the shaft, may be mentioned the fact that it is sometimes necessary to give the stroke-lines of the piston units of a large vertical engine a slight inclination toward each other, in order to make the connecting-rods bear squarely upon the crank-pins—this applying to an engine like Fig. 218 or 219, with a heavy rotor between rather widely-spaced bearings.

(c) **BEARINGS FOR STATIONARY ENGINES.**—In connection with the shaft we shall consider now the detail of the part of the framework upon which it rests. Fig. 319 shows typical examples of bearings from small high-speed engines. In I. there is a one-piece bushing of cast iron, lined with babbitt (detail of lining not shown);

... turning the bearing together with its ... screwing the cap down upon it. With ... between frame and bushing, the ... freedom that would be given by a ball ... plentiful lubrication, a bearing of ... away; but should anything happen to ... bushing in case taking off the wheel.



High-speed Stationary Engines. I. Detail from ... in two parts; III. Ring-oiling bearing in

... bearing is shown at II. The main ... at the right, while that on top obviates ... of the cap on the bearing-boxes. With ... movement of the shaft, the lower box can ... or repair. Beneath the main ... key which fits into a round hole ... in the bottom box, so as to hold the ...

... bearing with oil-rings is drawn ... is backed by a face ... surface is ... the level ... in view D are

shown the holes for lifting screws which are tapped in the several parts of the bearings. In the middle of the bearing, at the top, is a screw-pin which serves as a dowel, to keep the boxes from ever turning with the shaft. View C is a plan of the casing, with the boxes removed. The lubrication arrangements, besides the oil-well and the rings with peep-holes above them, include a light collar fastened upon the shaft at the outer end of the bearing, to catch all oil that escapes and return it to the well. At the inner end the oil drips upon the projection L, from which it is scraped by a little catcher on the crank-disk, and carried to the crank-pin.

The engine-frames in Figs. 237 and 238 have even simpler arrangements than those just shown. In Fig. 238 the babbitt-metal lining is cast right in the frame and the main cap. Fig. 237 has thin loose shells of babbitt, held just as in the connecting-rod, Fig. 305 I.

*Corliss-engine Bearings.*—In horizontal engines of the Corliss type, the bearings are usually made in four parts, with side-adjustment either by set-screws or by wedges. Fig. 320 I. shows vertical adjustment also, by means of the wedge drawn in detail at C. It is hardly possible that a lifting force greater than the weight of the shaft and wheel will ever be developed in an engine of this class; consequently, the top box is made light, and is held down simply by contact with the cap at the ends. The big hollow cap is characteristic. In view B, the upper half is a plan of the bearing-cap, and the lower half is partly a top view of the base, partly a section by a plane through the shaft-axis.

Fig. 320 II. shows a double wedge-adjustment, the wedges being drawn up by long studs, with nuts on top of the cap. With this arrangement in only one of the two bearings, it is possible always to square the shaft with the stroke-line, besides taking up wear—provided the bottom box can move, as in this case. Where the lower part of the bearing is solid with the frame, as at III., the wedges can be used only to adjust the side boxes to a proper fit. These wedges are raised by set-screws, which go through them and rest upon the seatings beneath them, so that the cap can be taken off without affecting the adjustment.

The bearing for a very large engine is given in Fig. 321. We

of the bearing—support, which insures a uniform distribution of pressure between journal and bearing. The main bearing is shown in the manner best shown by the developed section at D, in order to permit the circulation of cool-  
 water, which enters by a pipe following the path from P. By

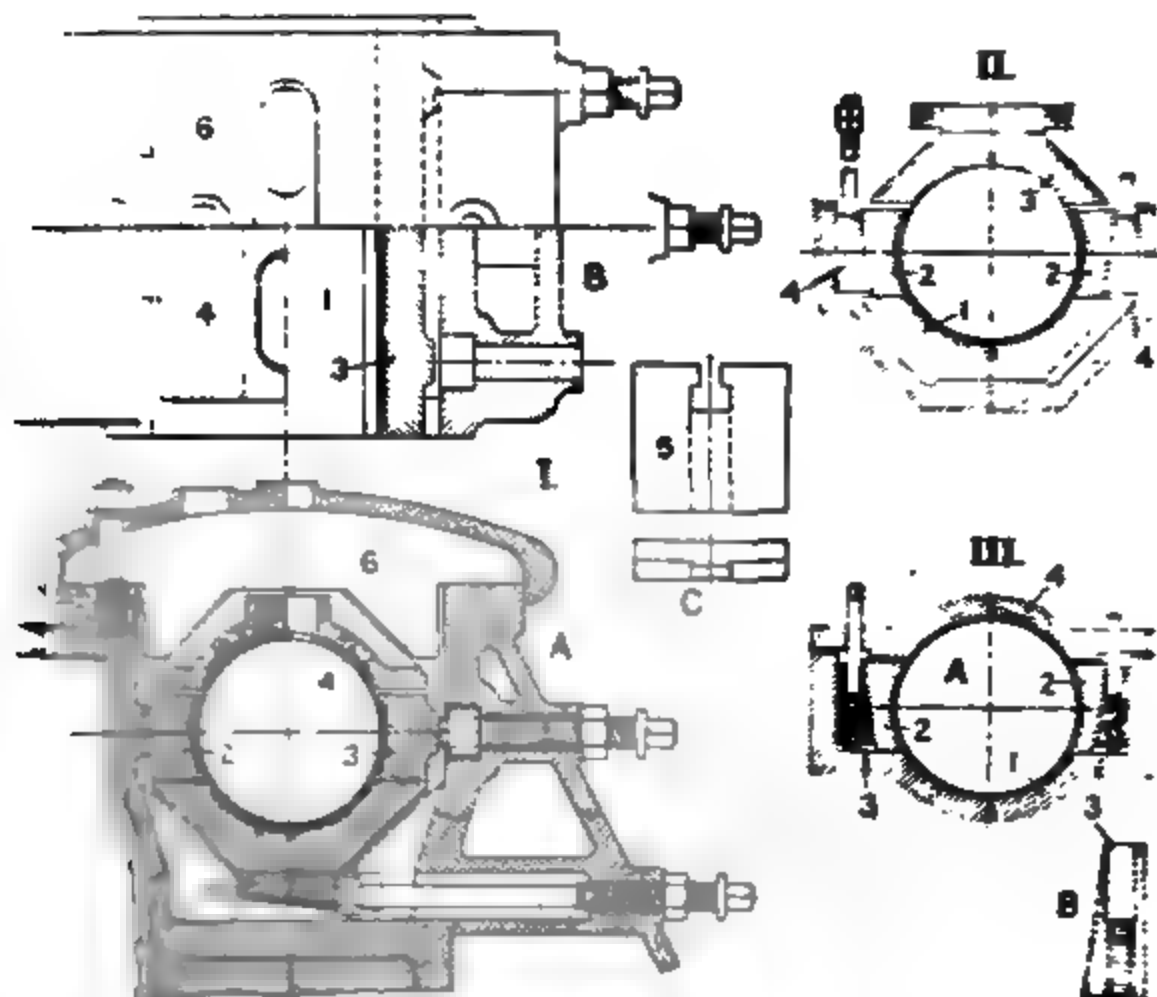


FIG. 20—Bearings for Engines of the Corlies Type. I. For frame in Fig. 19, scale 1 to 24; II, III. Different arrangements of the bearings.

making the water-block 5 a separate piece, the box 1 is given such a form that it can be turned freely on the shaft when the latter is raised up high enough to take its weight off the box—suitable handles or screw-handles being provided. In spite of the fact that the pressure element of the engine is vertical, the free lifting force must be up-stroke, over and above the weight of the shaft and a small comparatively small: so that the top box 3 is narrow and light, and is held down only by three long set-screws, the whole being suspended by a couple of lighter tap-bolts. This

Drawing shows the form of the frame-casting around the bearing, and also illustrates a detail of practice which is frequently followed in large engines, in that the bolts for the bearing caps come clear

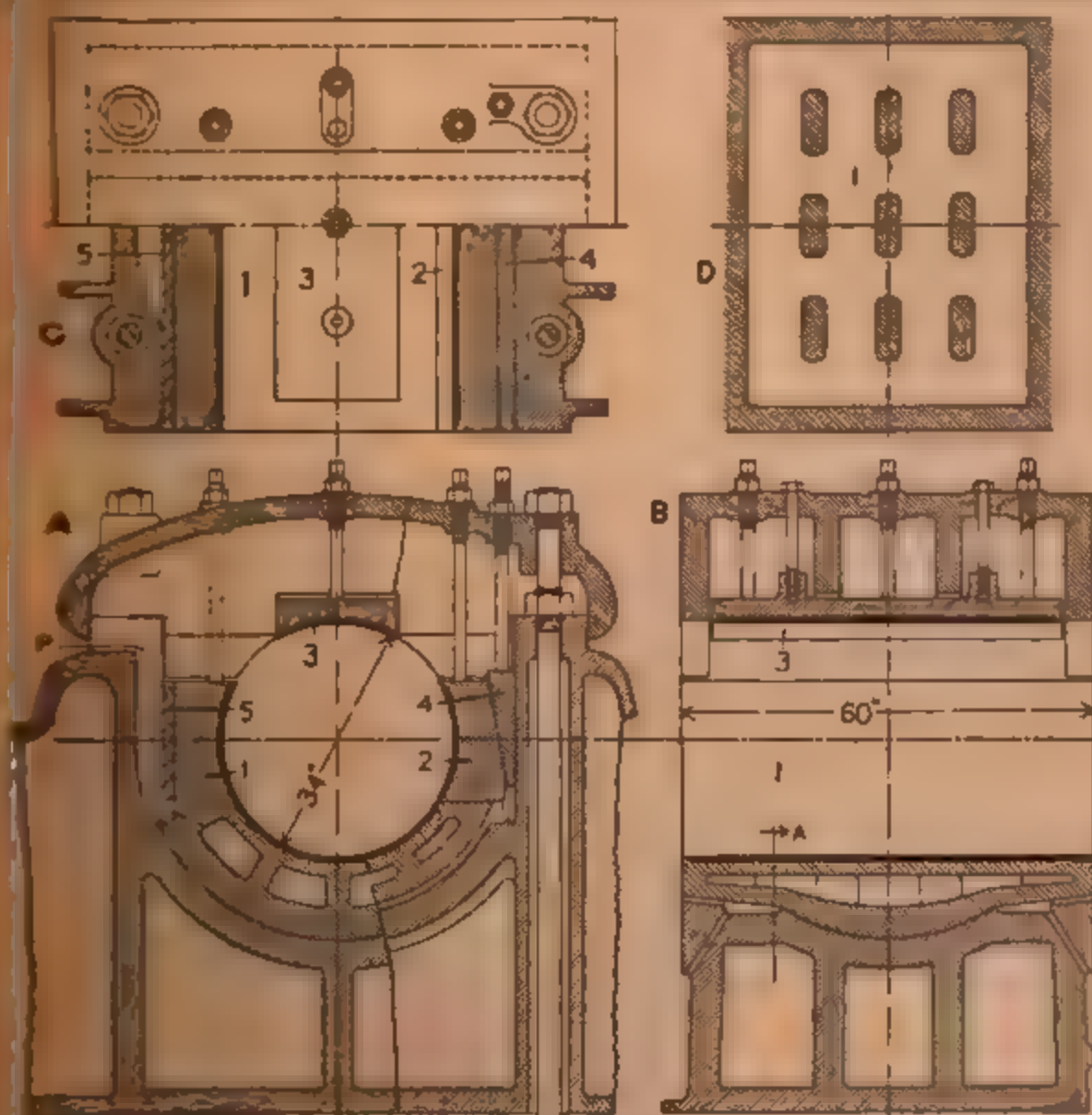


FIG. 321 — Bearing for Engine in Fig. 219 and Shaft in Fig. 312 II.  
Scale 1 to 40.

through from the foundation, taking their share in the duty of holding the whole engine in place. At the left of view B is seen the beginning of the light floor, a part of the main casting, which runs under the crank-pit to keep oil and water off the foundation.

(j) BEARINGS FOR MARINE ENGINES.—A fair idea of the form of these bearings can be got from the general views, Figs. 232 to 235, and from the frame drawings, Figs. 243 to 245. Two examples are given in detail by Figs. 322 and 323. The first has a square

bottom box of cast iron and a hollow cast-steel cap, both arranged for the circulation of water. The bolts are long studs, with special

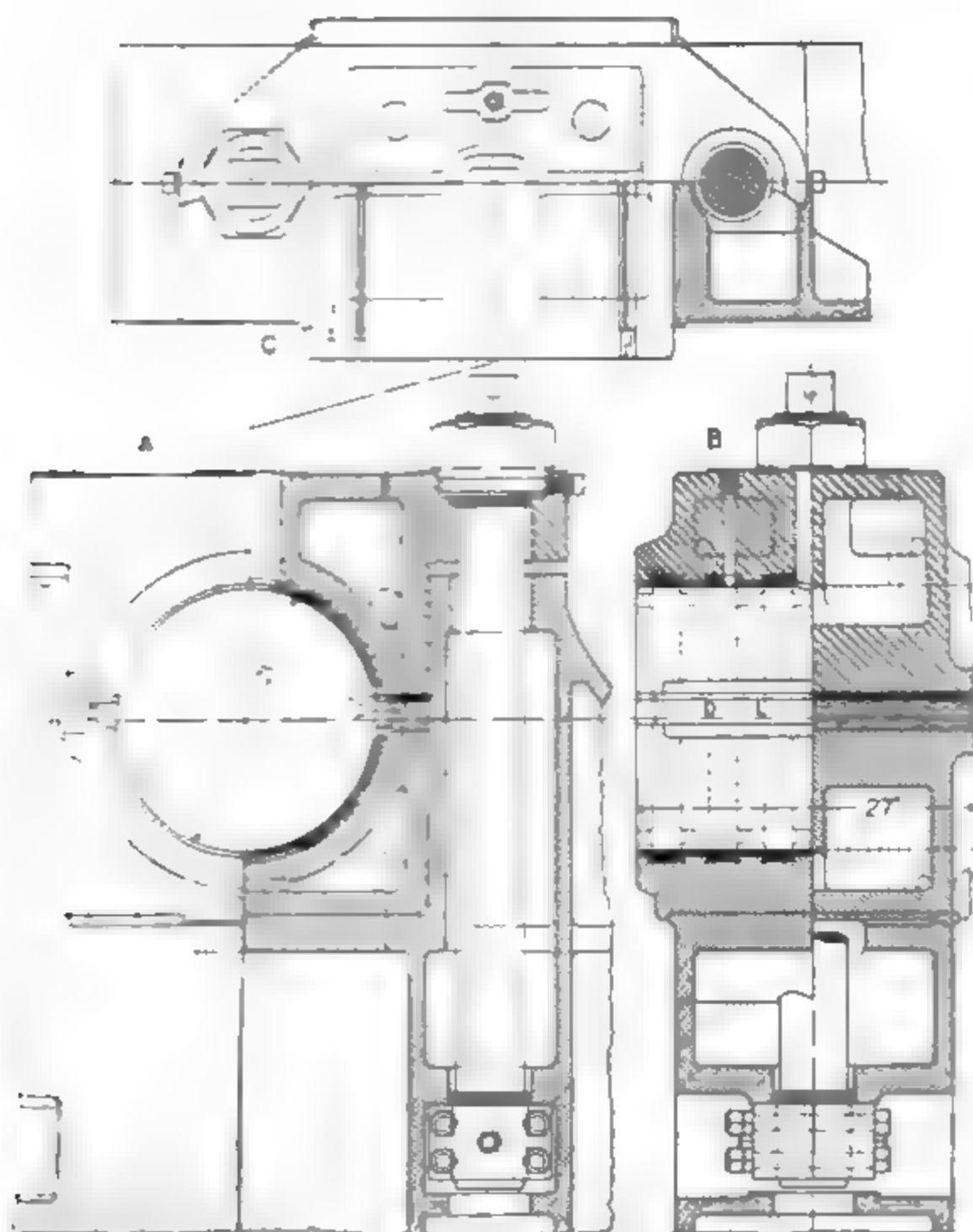


FIG. 322.—Bearing from Fig. 232, 21" shaft. Scale 1 to 20.

stud bolts at the bottom: only occasionally are ordinary studs, screwed into the frame, used for holding down the cap. The two halves of the bearing are separated by a distance-piece and a group of washers. The latter, with a total thickness of  $\frac{1}{2}$  in., is made



up of a graded series as follows: one each of  $\frac{1}{4}$ ,  $\frac{1}{8}$ ,  $\frac{1}{16}$ ,  $\frac{1}{32}$ , and two of  $\frac{1}{64}$  in.: so that it can be adjusted by 64ths to any thickness desired. These liners are held in position by dowel-pins, in the usual manner. In view C, the distance-block is in place at the left, but at the right it is removed. In B the upper edges of the dovetail ridges which hold the white-metal are dotted in outline.

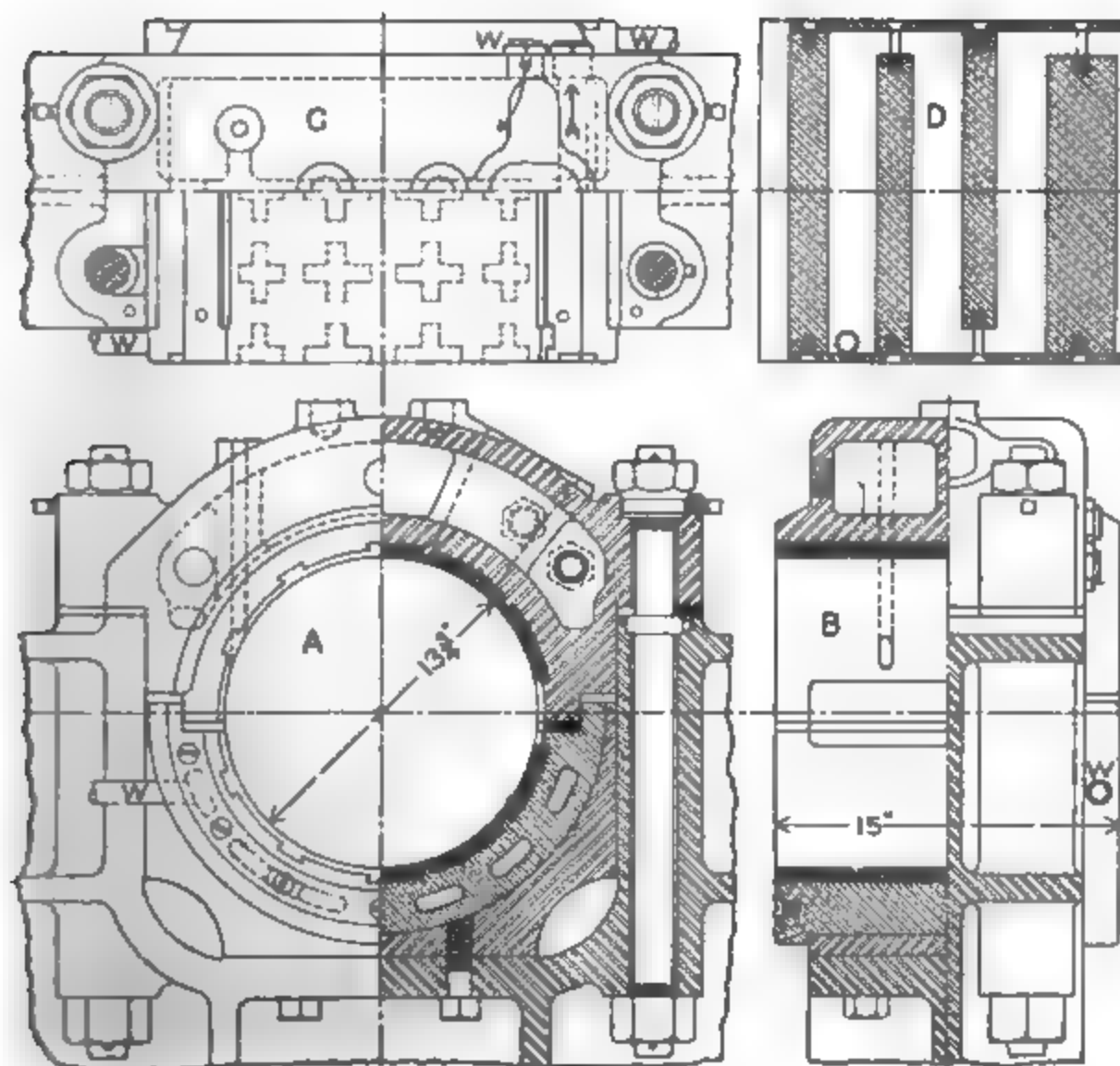
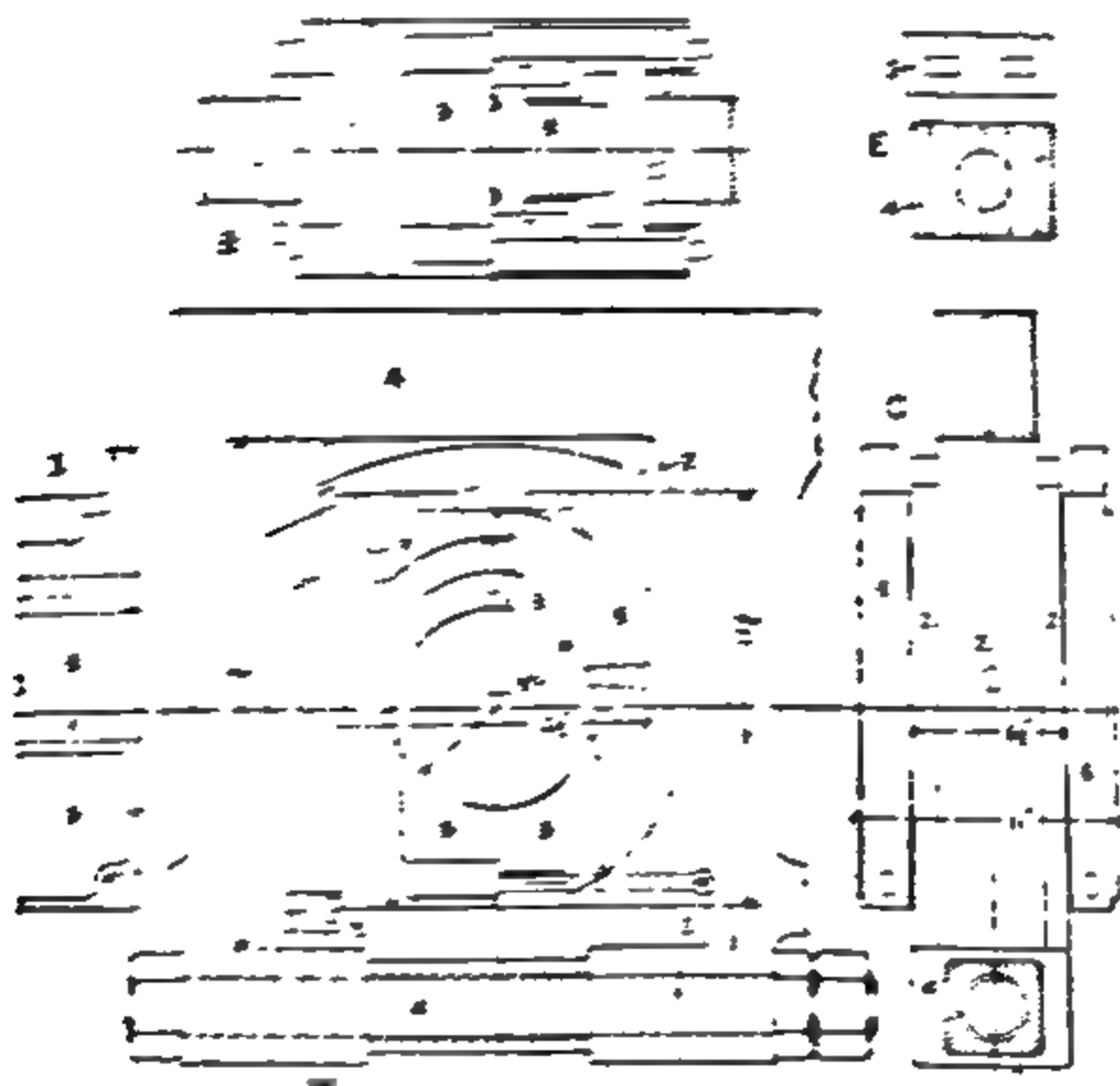


FIG. 323.—Bearing for U. S. Battleship "Mississippi"; compare Figs. 235, 295, 299, 309. Scale 1 to 12.

Fig. 323 shows a bearing in which the lower box is cylindrical, as in Fig. 244 also: but here this box rests in a "false" seating set into a square recess in the frame, instead of having a cylindrical seating formed right in the frame. The only apparent advantage is that the bearings can be adjusted vertically by putting liners

[illegible]

of the vessel is very important. Inside the two small oil-holes  
through the main cap there is a hole for passage of water, which  
will run to the bottom of the tank. The water passages for this cap are  
shown in the drawing. A 1/2 inch pipe runs one half and a pipe pro-  
jecting through the main part of the cap into the other half, to serve  
as a drain.

(g) **LOCOMOTIVE BEARINGS.**—A typical locomotive axle-box is given in Fig. 324. Inside the frame-jaw 1 there are two shoes, of which 3 is in the form of a wedge, to adjust the space for the axle-box 6. The enclosure is completed by the pedestal-brace 4, with a heavy through-bolt. The box 6 is here of cast iron, but is now much oftener made of cast steel. Into it is forced the bronze bearing 8, only a top bearing being needed in this case. The circle is completed by the light cast-iron "cellar-box," which is filled with wool-waste soaked with oil. Special details, belonging to the carrying function of the axle rather than to its service as the engine-shaft, are, the pocket 7 in the top of 6, wherein rests the support for the spring-rigging; and the manner in which the flanges on 6 are opened out toward the ends in view C, so that one end of the axle can move up and down in the frame independently of the other.

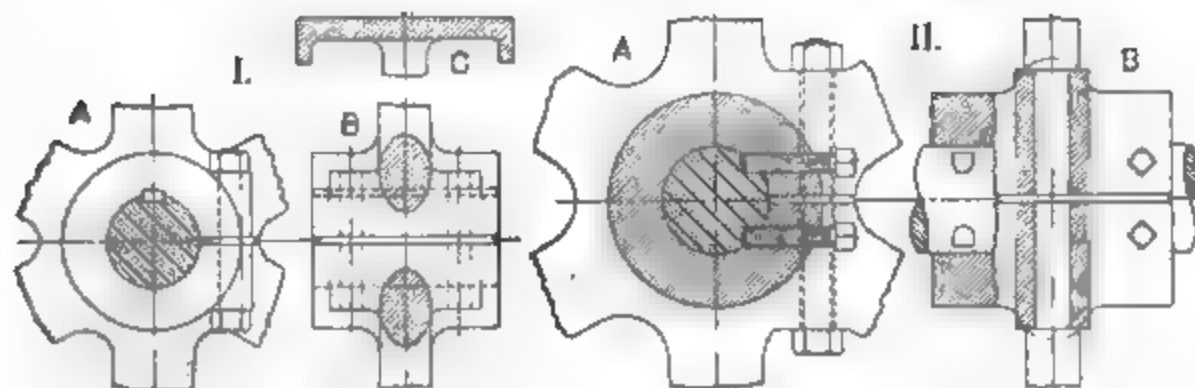


FIG. 325.—Fly-wheel Hubs for Small Engines. I. Standard design; II. For 15'' by 14'' engine, Figs. 237, 315, scale 1 to 16.

(h) **SMALL FLY-WHEELS.**—The general form of the wheel is very well shown by the illustrations in the first part of the chapter. Stationary engines for general service have wheels of the belt-pulley type; and this generally holds true, for the smaller sizes, even when the engines are direct-connected to generators, as in Figs. 202, 203, 206. For larger engines, direct-connected or directly loaded, the balance-wheel type with rectangular cross-section of the rim is usual, as in Figs. 209, 214, 216, 223, etc.

Small belt-pulley wheels are made with inside flanges on the rim, as shown at C in Fig. 325 I. In diameters less than 9 ft. they are usually cast in one piece; but very generally the hubs are split, on one side or all the way through, so that they can be clamped upon the shaft. Fig. 325 I. is a typical arrangement, with two

bolts at one side of the hub, and a common rectangular key. Fig. 325 II. shows a single bolt in the plane of the arms, and has six screws fitting into pockets milled in the shaft.

(1) **WHEELS IN HALVES.**—Wheels up to 16 ft. in diameter are commonly made in halves—the size of the largest piece that can be shipped on an ordinary railroad-car having a good deal of influence in this matter. One example of the balance-wheel type is detailed in Fig. 326 I. The hub is strongly clamped upon the shaft by the four heavy bolts; the strongest part of the rim-joint consists of the two I-shaped shrink-bolts or links. These are machined exactly to length between the heads, and the seating in the wheel-rim are similarly finished. The link is made shorter than the space it is to occupy by from one in one thousand to one in eight hundred. With a coefficient of elasticity of 30,000,000 lbs. per sq. in., a deformation of 1 in 1000 will produce, or be produced by, a stress of 30,000 lbs. per sq. in.: in this case the deformation will be mostly concentrated in the link, which is much smaller in cross-section than the body of cast iron which clamps together that nearly the full stress will be developed.

Wheels of this size will be completely finished in the shop; and to insure a neat fit at the joints, little screw-dowels are put into holes drilled and tapped half and half in the two parts. In erecting the engine, these will be put in first, along with the bolts through the lugs inside the rim, and the shrink-links are put in last of all.

Another form of connecting-link is shown in Fig. 326 II., the name “link” being here closely descriptive. At III. is a special joint, belonging to the engine in Fig. 217. The same U-shape tie, without the I links, is used in Fig. 203, two at each joint. These can be put in hot, so that they will have a good grip when cold: the temperature required for a shrink-fit of the degree above described is not very high, but is well below the beginning of “red heat.”

The tie-bar with keys, Fig. 326 IV., is sometimes used, Fig. 223 being an instance. This is not a shrunk joint, but is tightened by making the keys with a slight taper and driving them in hard.

Large wheels of the belt-pulley type are joined by bolted flanges after the manner of the sketch at V. In very large diameters these

wheels will be made in a number of segments, each with one or two arms, and with an arrangement at the hub similar to that in Fig. 327. Wheels have been built with the arms separate from

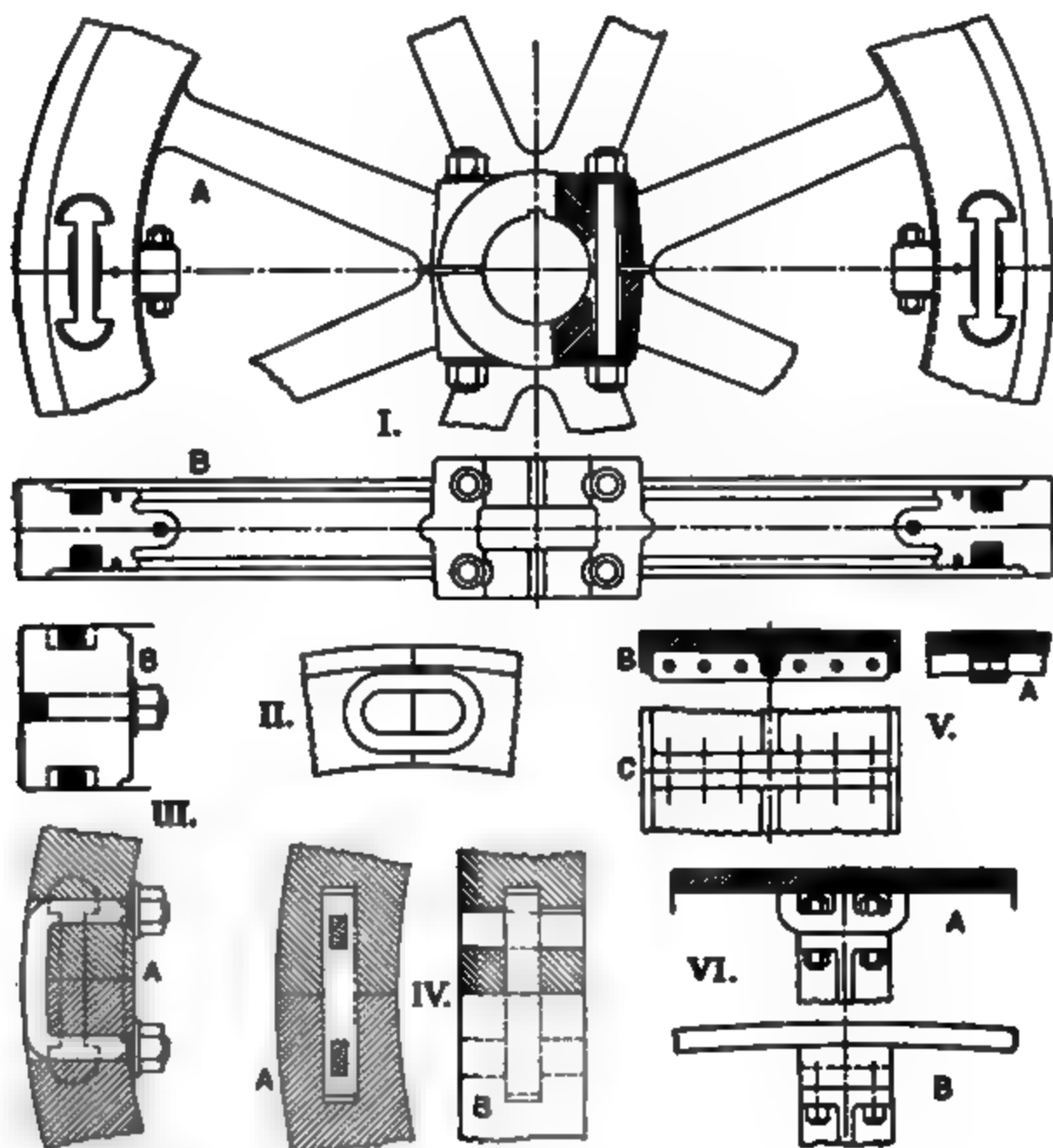
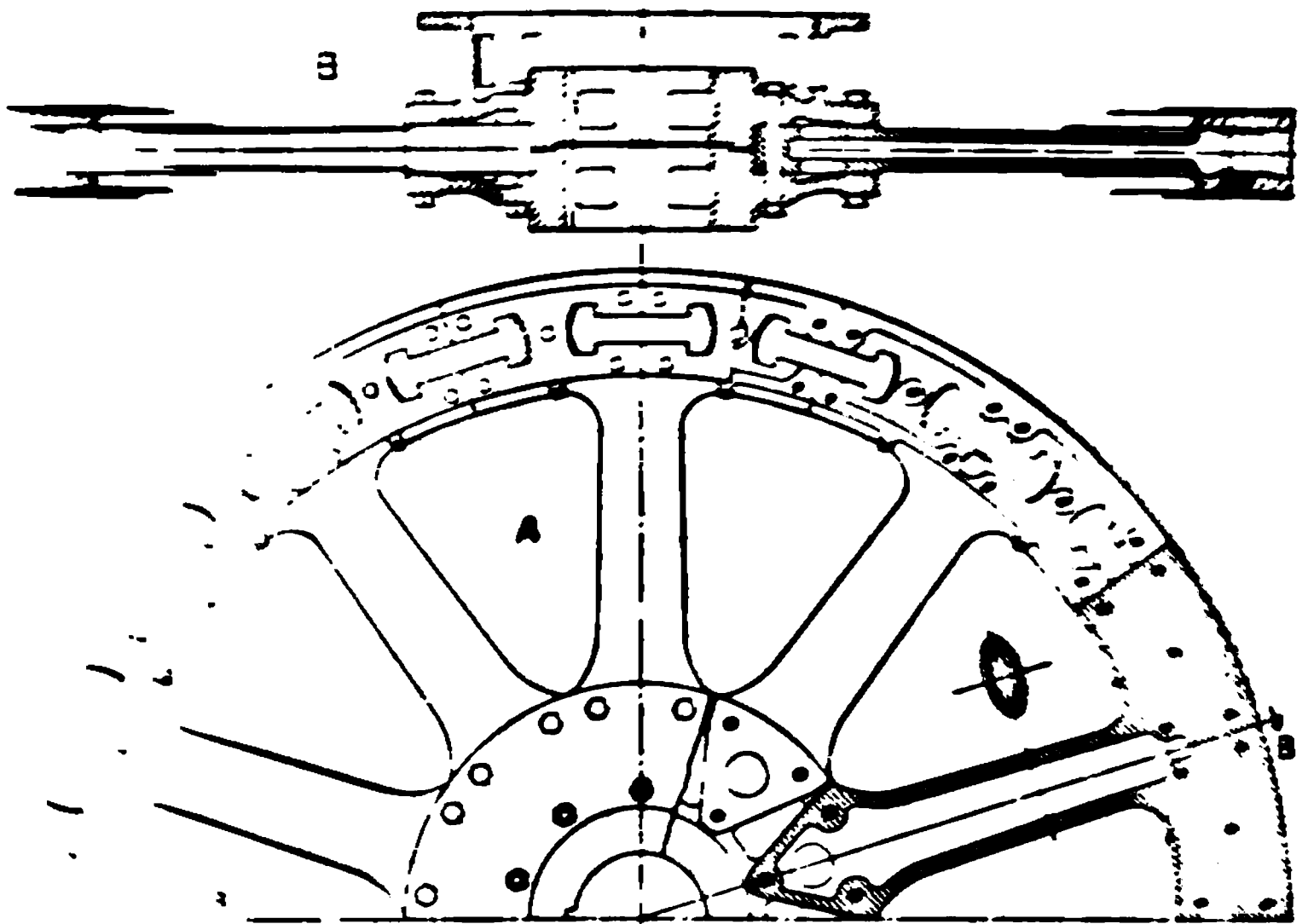


FIG. 326.—Balance-wheels of Medium Size. I. 16-ft. wheel for 26" by 48" Corliss engine, shaft in Fig. 312 I., weight 62,000 lbs., scale 1 to 48. II. to VI. Details of wheel-joints.

the rim-segments and bolted fast as in Fig. 326 VI.—the arms having the cross-shaped section seen in Fig. 210; but it is now standard practice to cast rim and arms together. A belt-pulley wheel in halves is seen in Fig. 211.

## 1. CONSTRUCTION OF THE ENGINE FOR VIL

**WHEELS IN SEGMENTS.**—Large and heavy wheels are usually constructed at the mill where they are used. In this design we note that the rim is in plates, the plates are perpendicular to the axis of the shaft. Between the plates on these plates the inner ends of the arms are firmly fastened by self-drilled heavy bolts. The rim consists of the rim wheel is one arm with segments of the rim the rim is cast iron. The rim joints are made with heavy plates. The center is reinforced by side plates of cast steel. The rim covers the angle of two arms. The rim is made in two sides so that there is nowhere more than one



Heavy-eight-foot Wheel for vertical Corliss engine, 42" and 86" by 60". Scale 1 to 84.

cross-section of the rim. The whole rim is strongly reinforced by stout pins, which are forced into reamed holes under pressure, then riveted cold.

For circumferential strength against centrifugal force, the divisions of the rim is a self-supporting ring, by its own I bolts: but besides this, through the

riveting of the sections together, the excess strength of two, between their joints, compensates for the loss of strength of the one which is reduced in section at its joint; so that the structure as a whole has a higher efficiency than any one of its circular elements. Here "efficiency" has the same meaning as in the discussion of riveted joints, being the ratio of the least effective cross-section to the full section of the rim.

Sometimes these large wheels are made with a core or center to the rim, as here, and side rings built up with several layers of rolled plate. The segments of these rings are simply butted together, without tie-links, so that circumferential strength is wholly a matter of shear upon the through-pins. Further, to avoid having several joints in the same cross-section, either the segments must subtend a large angle, or there must be a great many rivets so that the joints can be separated by very small angles. Finally, the rolled plates are likely not to be true enough, in thickness or in flatness, to make a really neat job without being planed. For these reasons, the method in Fig. 327 seems to be generally preferred. Quite often, however, the arms and rim-center are of cast steel, the arms being then cast solid; and the unit is frequently two arms with their share of the rim.

(k) STRAINS IN FLY-WHEEL.—In general, there are two major force-actions in fly-wheels, due to centrifugal force and to angular inertia. The resistance of the rim to radial force, considering it as a simple ring, has been discussed in § 36 (i); where the fact was brought out that the circumferential stress depends simply and only upon the linear speed of the rim. It is easy to remember that this speed is, in usual practice, somewhere near a mile a minute or 90 ft. per second. In the ordinary wheel, the rim is held in by the arms, as well as by its own tensile strength; to some extent this action is helpful; but it also favors the development of bending stress in the rim, since the parts between the arms now have some tendency to bulge outward. This last action is of more importance in broad, thin belt-pulley rims than in the balance-wheel type.

Whatever turning moment is transmitted from the shaft to the wheel-rim is carried through the medium of bending stresses in

the arms. This moment will be the whole when the engine is belt-loaded; with other unbalanced moment acts upon the wheel, with the hub, as in Figs. 325 and 326, it is fixed at both ends. When bolted at the must be considered as having more or less unless the bolts are very strong and very up hard. If there is freedom at the held against bending mostly or altogether reason why large belt-wheels should rim, to resist a bending load, and exemplified by Fig. 326 VI, is not a

An important incidental advantage a number of pieces is, that they strains, which are likely to be very castings.

Occasionally large wheels are shapes, riveted or bolted together tion belongs rather to large wheels.

Of the accidents that occur of the fly-wheel comes next frequency and as to the severity to defect or failure of the apparatus will be discussed in

(I) AIR-RESISTANCE TO W now always made with arms little air-resistance as possible large wheel at high speed is have gone so far as to enclose sheet-metal, from rim to hub churning air.

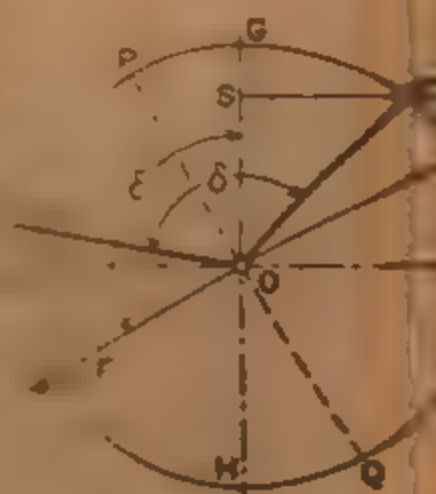


Fig. 326 VI. The Resilient Diagram.

Then Gill takes up with the crank;

If measured up right, or plus; if down, or left, or minus. movement.

inner or polar diagram a new way of representing of the driving or

and radial-center position or as its diameter the arm is equal the always equal, by the figure, or a few



## CHAPTER IX.

### VALVE-GEARS AND THEIR ACTION.

#### § 46. The Simple Eccentric-driven Valve.

(a) VALVE-MOVEMENT. — The general form of this simplest type of valve-gear is sufficiently illustrated in § 3; and the description of its action there given will serve as an introduction to the discussion now to be taken up. Our first task will be to develop convenient geometrical methods for showing the movement of the valve, or for determining its position corresponding to any position of the crank or of the piston.

The eccentric-rod is usually so long, in comparison with the radius of the eccentric, that the valve receives practically harmonic motion: then, for kinematic study, the whole mechanism, including the crank, may be "reduced" to the form shown in Fig. 331. This is a modification of Fig. 106; and by placing the slide beneath the eccentric-circle we emphasize the fact that the distance of the valve from its mid-position, or MV, is the same as the distance SE of the eccentric-center E from the vertical center-line SO. The position of the valve is defined by giving this distance, measured to the right or left: we shall call it the valve-travel, and denote it by  $t$ .

Now the movement of the valve is determined by the rotating eccentric (really a small crank), just as that of the piston is determined by the crank: and the relations as to velocity and acceleration derived in § 33 (b) and (c) apply equally well to the valve-slide, if  $\alpha$  is taken to be the position-angle of the eccentric, measured from the left-hand dead-center.

(b) MOVEMENT DIAGRAMS.—In corresponding to any crank-position  $\delta$  or COE in Fig. 332, we must find the length of ES or  $t$ . But a give  $t$  directly from the crank-angle repeatedly laying off  $\delta$ . One diagram is derived in Fig. 332, where the line GH, the eccentric-radius

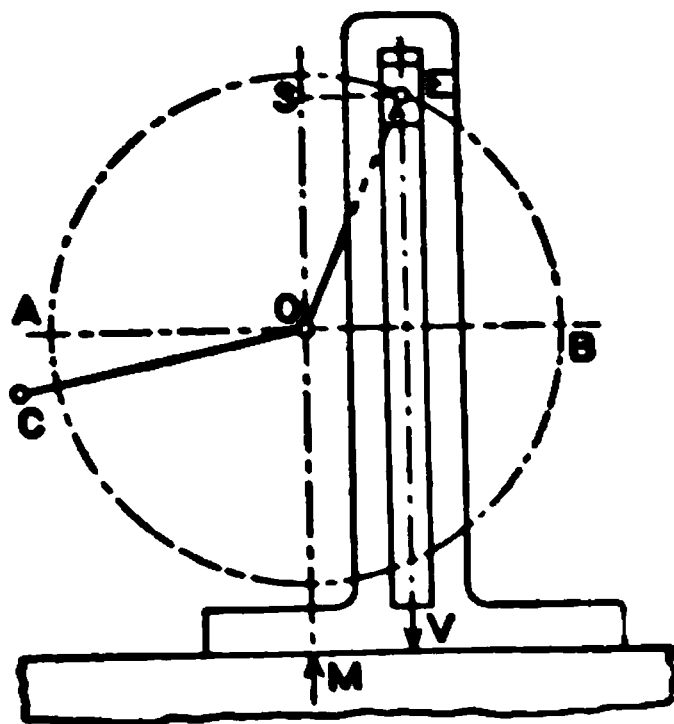


FIG. 331.—The Valve-mechanism. Reduced.

backward about O through constant position MN, where the perpendicular DF gives from MN, parallel to OP,  $t$  forward, or in the direction of rotation, is the Reuleaux or ordinate.

The derivation and for given in Fig. 333. We find the motion of the slip arm (here the eccentric-

On the line OB, which of the eccentric, draw the Then the intercept OF for the right-angled triangle also the sides OF and

is changing rapidly or

ms.—It is obvious that and dead-center OB—the crank will be perpendicular to the Reuleaux diagram, and will be the Zeuner diagram. Then the diagram would be: Draw the eccentric on its plus right-angles with the corresponding will give the base-line and Zeuner diagram, place the on the corresponding crank-circle on this radius as a center if we make the following construction, from which to estimate the piston is farthest from the crank when it is from mid-position the angle  $\delta$  be always measured in the direction of rotation of the crank. The engine stands or runs, the piston-stroke and valve-stroke are the same meanings of  $t$  as there

at the middle of each of the four diagrams of both kinds, and the valve is at mid-stroke and the values. In some of these diagrams other than those of Fig. 110, the same.

and  $\delta$ , draw a Reuleaux and a where the crank is when  $t = +\frac{1}{2}r$

DIAGRAM.—Having established the diagram and determining the move-

the general conditions is found in the diagram.

or how this valve, moving  
the steam-distribution.  
shown in mid-position on  
besides  $r$  and  $\delta$  as repre-

steam-lap;  
exhaust-lap.

left port, or for the head end  
two methods, are given at III.

In IV this line is tangent to the  
a zero-intercept—the valve is in mid-  
relatives the valve moves toward the  
positive and increasing in both figures:  
 $t = (Q_1 - c) \cdot s$ , the valve-edge and port-  
the port is just beginning to open. For  
later, the valve takes the position shown  
and the port is open by the small amount  $c$ ,  
ad. When the crank is at any position (OC,  
onds, we have  $t = CF = EO$ ; and it is evident  
port-opening is equal to  $(t - s)$ . In order to  
dtraction of  $s$  from  $t$ , we draw in III the lap-  
MN at the distance  $s$ ; and in IV., draw the  
with  $s$  as radius. Then the segment QDR and  
ed are identical diagrams of port-opening. We  
sion begins—or, we “have admission” at Q,  
ing is at D, while cut-off takes place at R. It is  
he determining of admission and cut-off is simply  
inding crank-positions for which the valve is at a  
ce from mid-position.  
crank passes R, the valve keeps on moving back  
ght—as is shown by a plus but decreasing  $t$  until  
mid-position again when the crank is at ON: then it  
d the left, and soon opens the exhaust-port, this occur-  
a  $t = -i$ . The beginning and end of exhaust, or “re-  
and “compression,” as also the port-opening during ex-

velocity of the ...  
slowly.

(c) Reuleaux  
if the ecc.  
when  $t$  w  
dicular to  
lie along  
a rule for  
an eccen  
dead-cent  
sponding  
determine  
eccentric  
line mea  
diameter  
assumpti  
crank-ar  
crank-sh  
toward th  
from crank  
shaft: \*  
and wh  
agree, the  
stated, b

PROB.  
quadrant  
show on  
where  $t$  h  
grams,  $t$   
as to pos

PROB.  
Zeuner  
and  $t =$   
method

\* A  
case 12

exhaust lap-line TS or the inside  
using only the plus valve-circle  
drawing another valve-circle  
intercept shows left-hand or minus  
not necessary, for it is evident  
by either intersection,  $e$  or  $e'$ .  
reason from the Zeuner diagram  
the intersections of valve-circle  
when, as is usual, only the one  
early in mind, not only that the  
distance for one of these events, but  
and which way it must be moving.  
circle alone, if we were to draw  
for the release-position, we should  
the valve is at the distance  $i$ , it is  
should be to the left and moving  
when the crank is at  $eOT$ . For these  
Reuleaux diagram is clearer and  
ver.

As valve there is over-travel; for if  
draw  $b$ , and draw VW parallel to ST,  
distance  $b$  from the lap-circle, we see  
enough fully to open the port.  
over-travel on the steam side: but  
of the time in single-valve gears  
maximum opening for admission is  
the port.

Other end of the cylinder, the events  
truly opposite to those shown, with  
laps are not equal, they must be drawn  
sections found. Generally, both sets  
in a Reuleaux diagram, dotting  
the Zeuner diagram, we usually  
and, for equal laps, the same

draw a complete  
the steam-distribution—espe-

compression—for both ends of  
of opening and for over-travel

—Having established simple  
between the positions of the valve

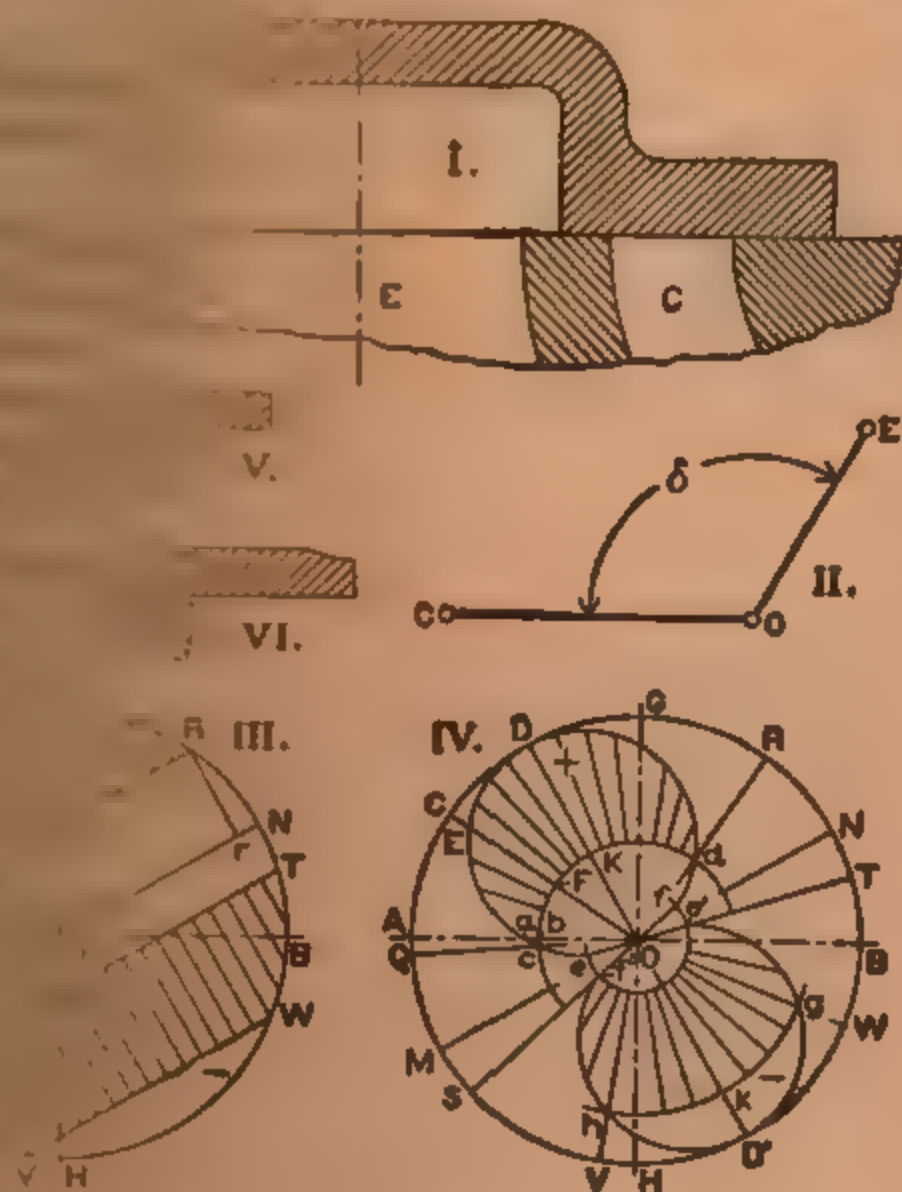


FIG. 334.—Valve-diagrams.

rank, our next step is to extend these to the valve  
The primary, determining diagrams are shown in  
the valve-diagram (of either form) is combined  
position diagram from Fig. 111: then  $DC'$  and  
simultaneous determinations. The distortion from sym-

With reference to the diagram in Fig. 235, it will be seen that the valve is represented by a circle, and the piston by a circle of smaller diameter. The valve is represented by a circle, and the piston by a circle of smaller diameter. The valve is represented by a circle, and the piston by a circle of smaller diameter.

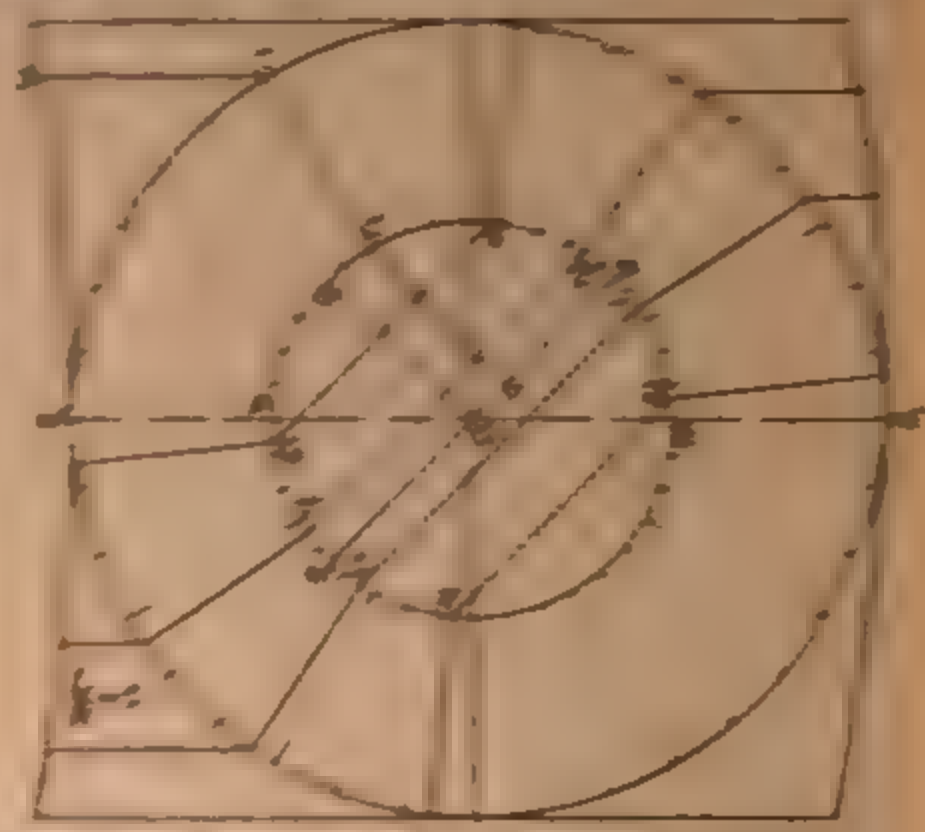


FIG. 235.—A diagram showing the relationship between the valve and the piston.

By this figure it will be seen that the valve is represented by a circle, and the piston by a circle of smaller diameter. The valve is represented by a circle, and the piston by a circle of smaller diameter. The valve is represented by a circle, and the piston by a circle of smaller diameter.



FIG. 335.—The Valve-ellipse.

The curve got by this method is elliptical in form, and harmonic motion for the piston as well as the valve it is

ellipse. The effect of the connecting-rod is here shown by dotting in parts of the simpler curve. The lap-lines are now drawn parallel to MN: and the four events, admission, cut-off, release, and compression, are located by the intersections marked A, B, C, and D, respectively. Dotted lines and primed letters are for the crank end of the cylinder.

In order to equalize the cut-offs, making them the same as with harmonic piston-movement, the laps would have to be changed to the values  $s_1$  and  $s_2$ , as marked on the figure. This would reduce the lead—which is shown by the distance from Q to the point where the curve is tangent to the end-line—almost to zero at the head end, while nearly doubling it at the crank end, besides changing the widths of the port-openings all through both admission-periods. A diagram of this type furnishes the most satisfactory data for a comparison of valve-action with realized steam-distribution, as shown by the indicator; and this matter of symmetry of action will be more fully discussed farther on.

For an autographic diagram, to be drawn by the engine and to serve as a test of the working of the valve-gear, especially with releasing gears of the Corliss and similar types, this elliptical curve is the most convenient, the motions necessary in the apparatus being the same as in the steam-engine indicator. With a device of greater complexity, and requiring careful adjustment, it is also possible to draw autographic diagrams of the Zeuner type.

Another way of representing the travel of the valve with reference to that of the piston is shown in Fig. 337, where two displacement-curves like Fig. 125 II. are laid out. Of course, the two movements are shown separately and are related through the crank-angles, as in Fig. 335; but the relation is simpler and more direct. Where the valve-travel must be found by laying out successive positions of a complex mechanism, instead of being got from a simple primary diagram as in Fig. 335, this last method collects the results into what is probably the most useful form.

Curve I. shows the travel of the piston along its stroke-line, the latter being represented by the whole ordinate between MO and NP; which is divided into ten equal parts by the horizontal ruling, for convenience in locating positions on the stroke. By

... on  $\Pi$ , we get the whole steam-distribution,  
 ... intersections having the same lettering as on Fig.  
 ... piston-positions are determined by draw-  
 ... through A, B, C, etc., to curve L  
 ... to a simple example the available methods  
 ... and representation, we will next take up a

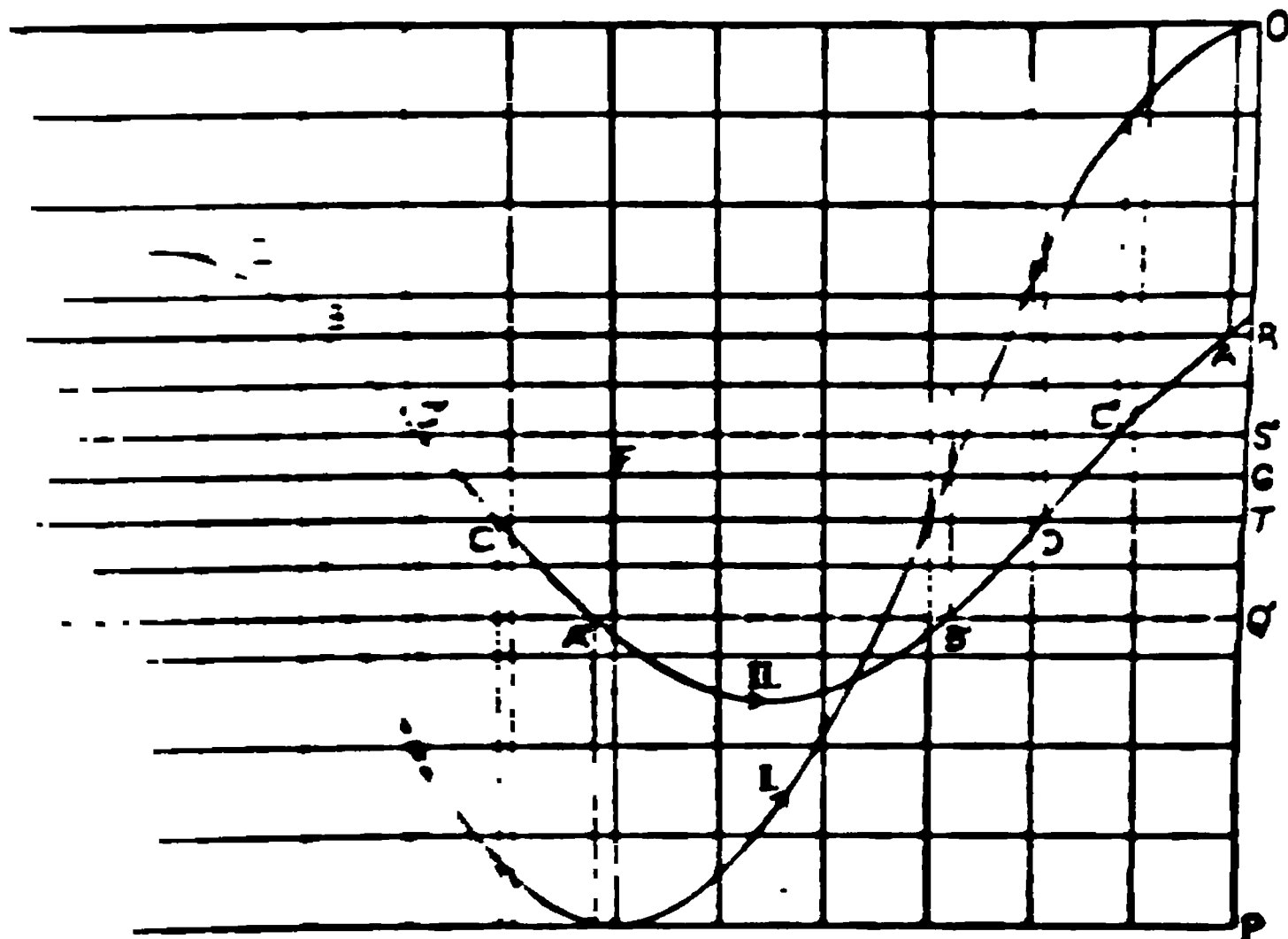


FIG. 335.—Developed Diagrams.

... principles which influence the form and action

... ANGLE OF ADVANCE.—The evolution of  
 ... in Fig. 335. The simplest possible  
 ... the valve just covers the port when  
 ... by an eccentric at right-angles to  
 ... coincides with the dead-center.  
 ... the very decided fault that the admission  
 ... during the entire half-revolution.  
 ... as shown by the Beuleaux dia-

... the first step is to give the



valve a lap, so that it will not uncover the port until the eccentric has turned through a certain angle from the vertical, and will close it at the same angular distance before the other mid-position, as in III. Along with this change, the eccentric must be advanced beyond the position at right-angles to the crank, so that when the latter is on dead-center the valve will be at a distance from its mid-position equal to the lap plus the lead. The effect upon the diagram is shown at IV.: in V. the valve is sketched;

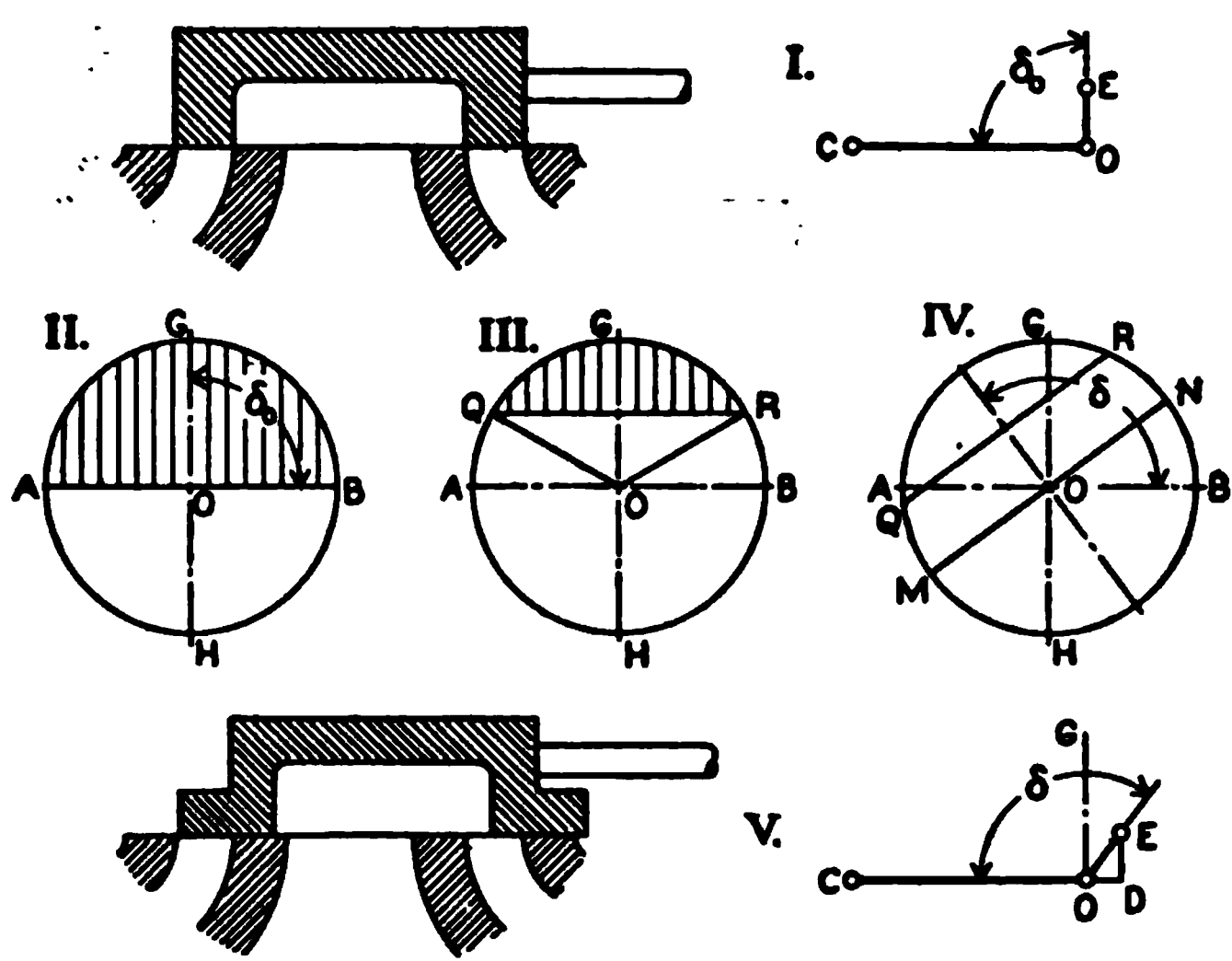


FIG. 338.—Evolution of the Valve.

and what is often called the angle of advance,  $\gamma = (\delta - \delta_0)$ , or  $(\delta - 90^\circ)$  in the usual engine, is determined by the relation

$$OD = s + e = r \sin (\delta - 90). \quad . \quad . \quad . \quad . \quad (311)$$

An important deduction from this figure is, that if the admission is to be very short, the lap must be very large relative to the radius, and the width of port-opening correspondingly small. This matter, together with the changes in the action of the valve on the exhaust side, will be considered when we come to valve-gears with a variable eccentric.

drawing in the  
the important  
336. The con-  
ing vertical

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of graph



Sta-  
of

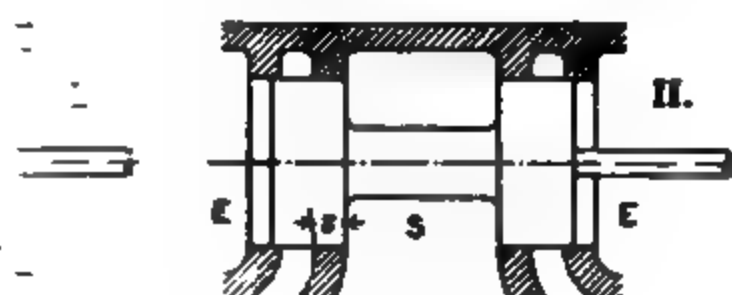
t

m

t

t

—There are two typical forms of  
piston form, and these are made  
Examples of both have been  
a more detailed description will  
distinction now to be drawn, and one  
present side of the subject, is illus-  
arrangement, having the live steam  
in the middle, as in the plain flat



Direct and Indirect Valves.

the first, with the steam in  
the ends, and with the laps inter-  
and indirect. Another way of defin-  
that they have respectively outside  
a general distinction, applying equally  
action valves, is expressed by stating  
the inward, moving toward the middle  
the valve opens outward—referring par-  
at either end. In other words, the  
by moving in the direction of the  
opening is a preparation, while the  
stroke and closes with it.

tion of the eccentric, due to a change  
valve, is shown in Fig. 340: the  
valve-travel distance is secured by  
a position diametrically opposite  
These are the characteristic eccen-  
of valves. In II. it is simpler to  
rather than to measure it all the

(h) EFFECT OF A REVERSING ROCKER-ARM.—Sometimes a rocker-arm pivoted at or near the middle is interposed between the eccentric and the valve, as is the case, for instance, in the

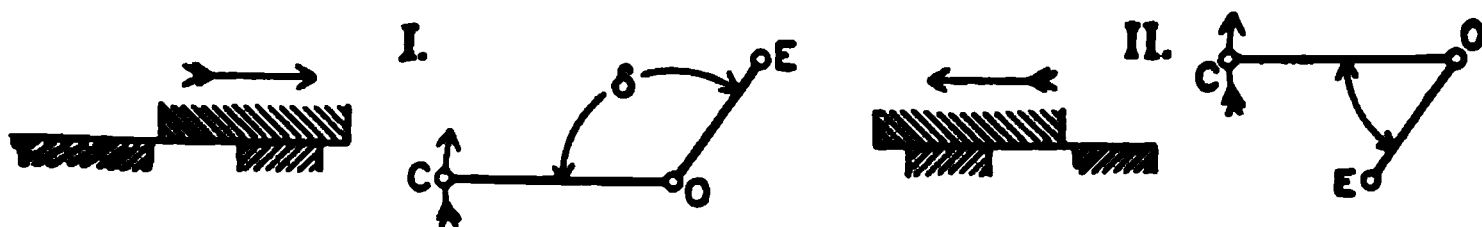


FIG. 340.—The Two Eccentric-settings.

engine shown in Figs 2 and 4. The result of this is brought out in Fig. 341, which is drawn for the locomotive, where the cylinder and axle interchange the characteristic positions for the stationary engine, as given in Fig. 110, but the zero dead-center is still taken at the left. To compensate for the reversal of motion by the rocker-arm, the eccentric must be diametrically reversed on the shaft; so that with a direct valve it has the setting proper

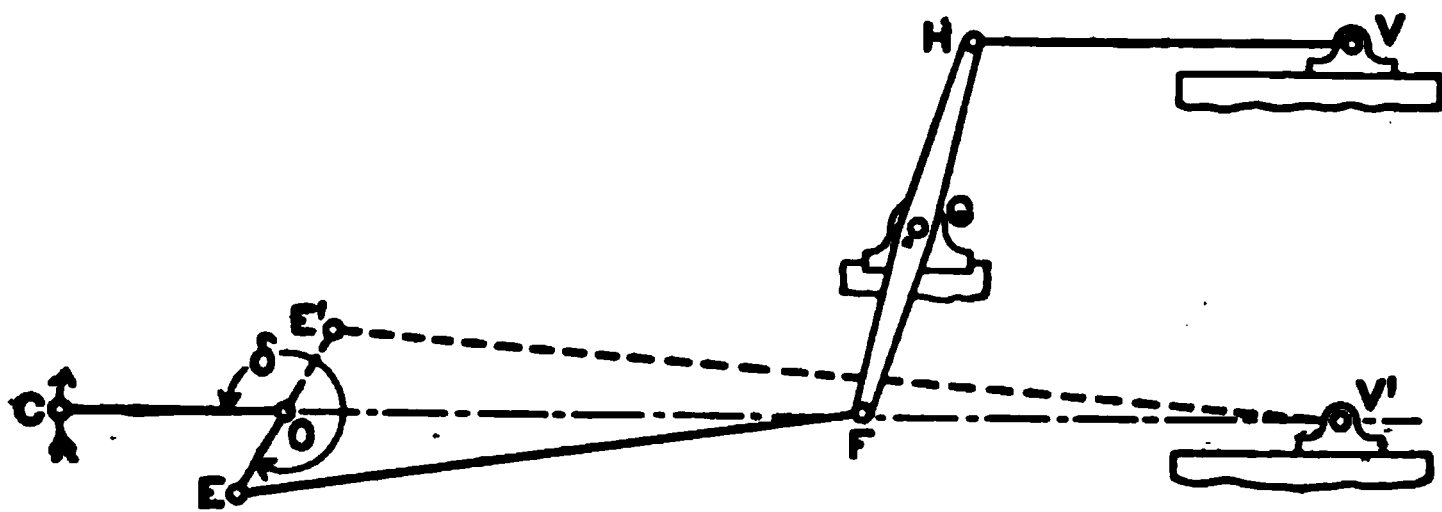


FIG. 341.—The Reversing Rocker-arm.

to the indirect, and vice versa. In this case, however, it is usual to disregard the double reversal in the mechanism, and to draw the valve-diagram as for a direct-connected eccentric, in the usual position. It is evidently from its analogy to this effect upon the eccentric-setting that the name "indirect valve" is derived.

(i) ENGINE WITH SEPARATE STROKE-LINES.—The occasional design in which the stroke-line  $A'B'$  of the valve makes an angle  $\beta$  with that of the piston is typified in Fig. 342. The primary position of the eccentric, for the conditions of Fig. 338 I. and II., is now  $OG'$ , perpendicular to  $A'B'$ . Following the usual rule

(g) To find the slide velocity with a given crank position, shown in Fig. 11, it can be found by the method having to be illustrated in Fig. 12 at the end of the chapter.



valve, as the motion is changed, by changing the crank position, and instead of well to that the of the particularly direct valve piston-rod indirect.

The from the necessary reversing to that tric-set estimat way

— If we add dead-center  $OB'$ , we get  $OD$ , which is shown at I.

— In Fig. 12, which also shows the characteristic diagram with an indirect valve. The angle  $\delta$  is now  $B'OD$ , instead of  $B'OD$ . Keeping the same in the matter of directions, this

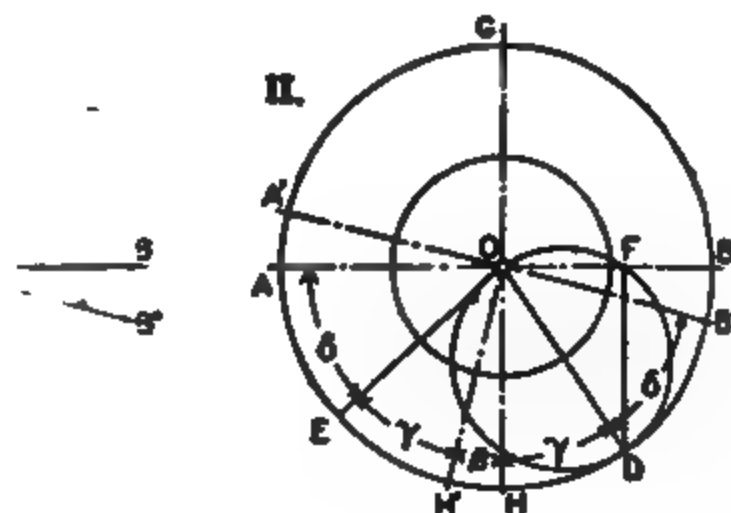


Diagram of Separate Stroke-lines.

— If we travel  $t = (s + e)$  for the crank at  $OA$ , and drawing the perpendicular  $FD$ , we get the angle  $\delta$  is now  $B'OD$ , instead of  $B'OD$ .

### Valve-diagrams: Valve-gear Problems.

— ELEMENTARY DIAGRAMS.—Besides the simple diagram 333, proofs with more of the usual elements will now be given. In both, a relation between the actual valve-travel, as measured distance, and the corresponding crank-position.

— In Fig. 13, whose side  $ES$  is  $t$ , is placed the crank  $OCF$ . When the crank was on its dead-center was at  $OE_0$ ; and, since  $COE_0$  and  $EOE_0$  must both be  $\alpha$ . The angle  $AOM = \gamma =$  the angle. This angle being constant,

the OCF-triangles will have their bases on the same line MN; a perpendicular upon this line from C, being always the other side of the triangle, will be equal to ES or  $t$ .

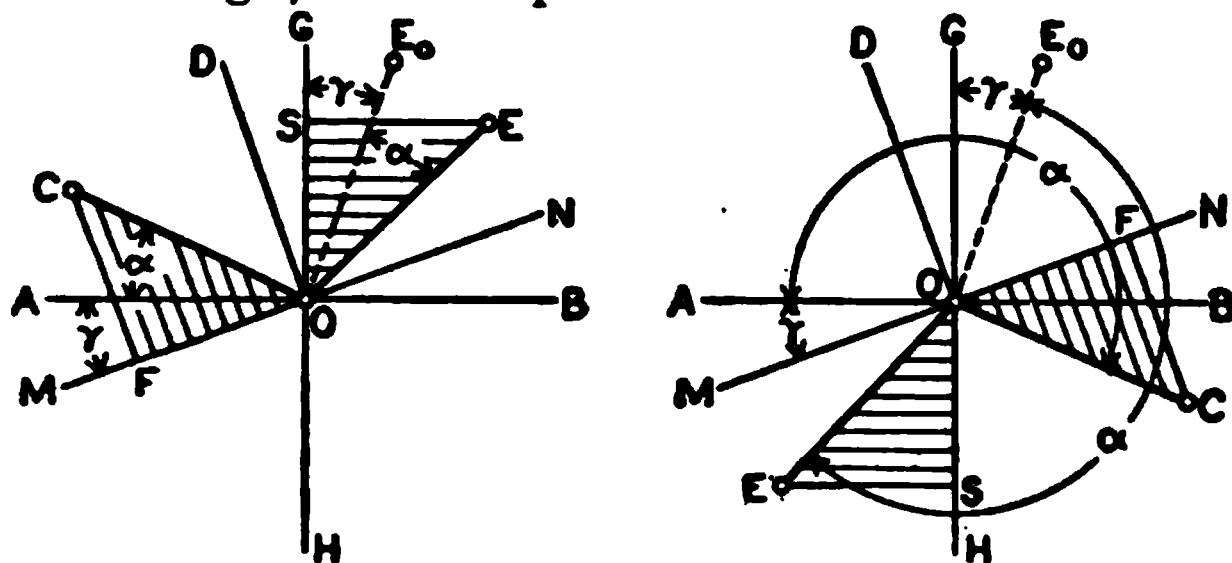


FIG. 343.—Proof for the Reuleaux Diagram.

For the Zeuner diagram, the triangle OTE, between the eccentric and the stroke line, is placed at OFD, with its side OT or  $t$  resting on the crank as OF. Then since  $\text{COE} = \delta$  and COD is the same as EOT, we get

$$(\delta - \text{COD}) + \text{EOT} = \text{DOT} = \delta:$$

which shows that no matter what the crank-angle AOC, the hypotenuse OD of the right-triangle will have a definite, constant posi-

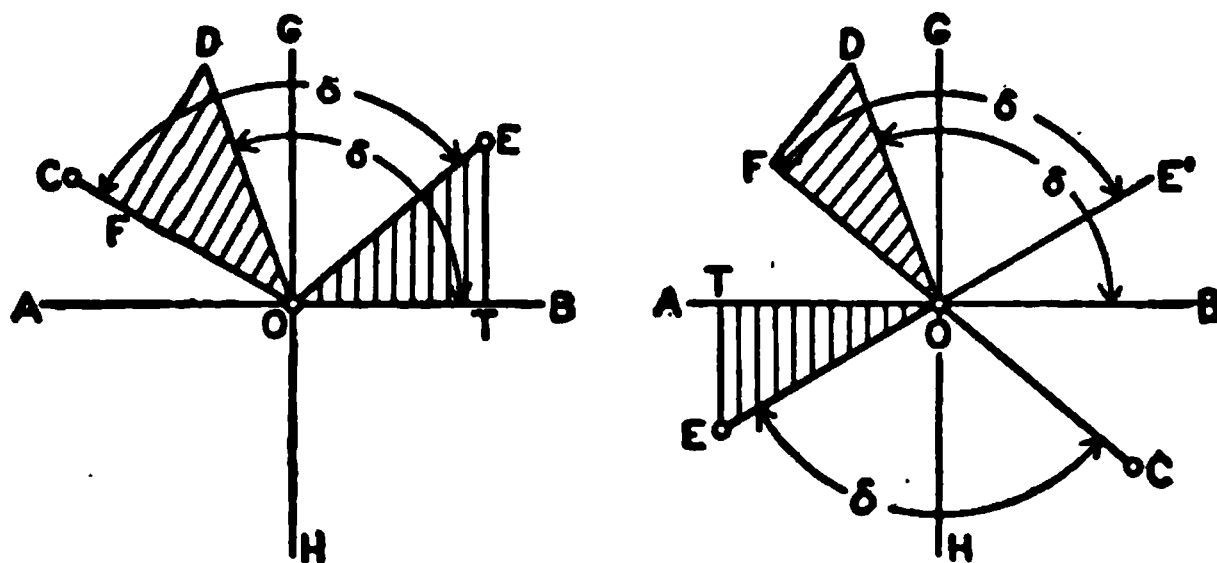


FIG. 344.—Proof for the Zeuner Diagram.

tion; and the apex F will then be on a circle of which OD is the diameter.

In these figures, the relations are also traced out for positions



line QR: self-evident on the Reuleaux diagram, this can be independently shown for the Zeuner from the equality of the right-triangles AUO, DFO.

Any event in the valve-action can be located by giving either the crank-angle at which it takes place, or the corresponding piston-travel with infinite connecting-rod. Thus the cut-off is fixed either by the angle AOR or by the ratio of AP to AB: but the admission-line OQ can be located only by the angle of lead, AOQ or  $\epsilon$ .

(c) PROBLEMS ON THE SIMPLE VALVE-GEAR.—The following résumé of the symbols used in this discussion will be found convenient:

$r$  = radius of eccentric, half of total travel of valve.

$l$  = length of eccentric-rod.

$\delta$  = eccentric-angle, measured from crank toward eccentric in direction of rotation.

$t$  = valve-travel, or distance from mid-position at any instant.

$s$  = steam-lap, outside on a direct valve, inside on an indirect.

$i$  = exhaust-lap.

$b$  = width of steam-port.

$e$  = lead, measured in port-opening.

$\epsilon$  = angle of lead, plus when measured from admission-line toward dead-centre.

A few practical problems will now be given, all having a direct bearing on valve-setting or design. Others can be devised, but many of them are useful only as illustrating the geometrical possibilities of the diagrams.

PROBLEM 4. Given  $r$ ,  $s$ , and  $\epsilon$ : find  $\delta$  and cut-off.

PROBLEM 5. Given  $r$ ,  $\epsilon$ , cut-off, release: find  $\delta$ , laps, and compression.

PROBLEM 6. Given  $r$ ,  $e$ , cut-off, and compression: find  $\delta$  and release.

PROBLEM 7. Given  $s$ ,  $e$ , and cut-off: find  $r$  and  $\delta$ .

(d) VALVE-SETTING.—The amount of adjustment possible after the engine has been designed and built varies with the type of valve-gear. In many single-valve engines, where the eccentric is carried by a shaft-governor or where it is keyed to the shaft, everything depends on its being correctly designed but sometimes the eccentric

can be rotated on the shaft, so as to change the angle  $\delta$ , and clamped in any desired position. In most engines the length of the valve-rod or of the eccentric-rod can be varied; and in the more complex valve-gears there are likely to be a number of points at which this kind of adjustment can be made.

The two conditions to be met are, first, that the valve-movement shall be symmetrical, so that the steam-distribution will be as nearly as possible the same for the two ends of the cylinder; second, that it shall be properly timed with reference to the motion of the piston.

With the engine cold and the valve-chest open, the two adjustments, of rod-length and of eccentric-angle, would be made together until the leads were equal and had the proper value—the engine being repeatedly placed first on one dead-center, then on the other. Or, if desired, equality of leads may be partly sacrificed to equality of cut-offs. With the help of the indicator, the valve can be set with the engine in running condition, stopping it for adjustment after each trial. This latter is, in many cases, the final method.

(e) CHANGE OF ROD-LENGTH.—The effect of this adjustment is shown in Fig. 346: the dotted lines show symmetrical working

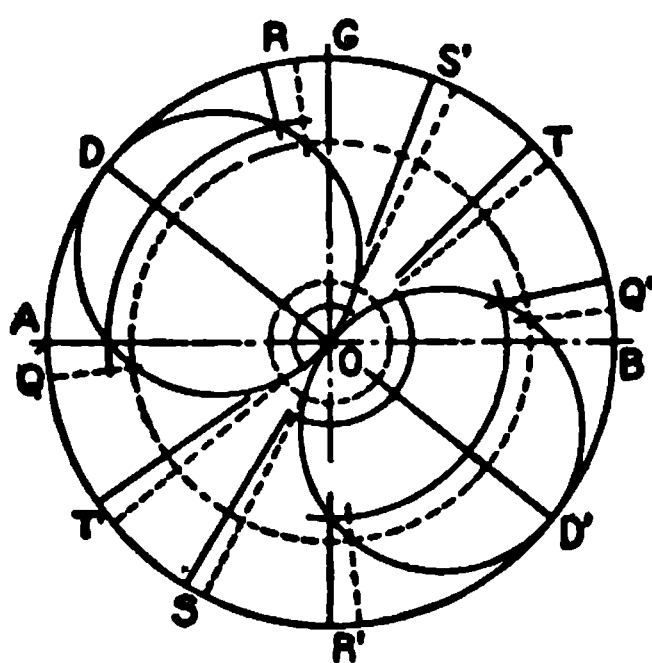


FIG 346 —Rod-length Changed. admission is shortened and exhaust lengthened; while the opposite effects are produced in the other end.

Adjustment under the indicator is illustrated in Fig. 347. As shown by I., there was quite an inequality in the cut-off and in the power developed in the two cylinder-ends: this could also be detected

or equal laps, the full lines the result of lengthening the rod, in a direct-valve engine; two circles being used so as to separate the indications for the two ends. Referring to Fig. 334 I., we see that to shift the mid-position to the left will increase  $s$  and  $i'$ , decrease  $i$  and  $s'$ —this notation distinguishing the ends just as does that used for the events on the diagram. Then for the head end, admission is shortened and exhaust lengthened; while the opposite effects



by the sound of the exhaust-puffs. Through uncertainty as to the type of valve, the rod-length was at first altered in the wrong direction, with the effect shown at II. Reversing this, and correcting in the proper direction, the symmetrical steam-distribution shown at III. was secured.

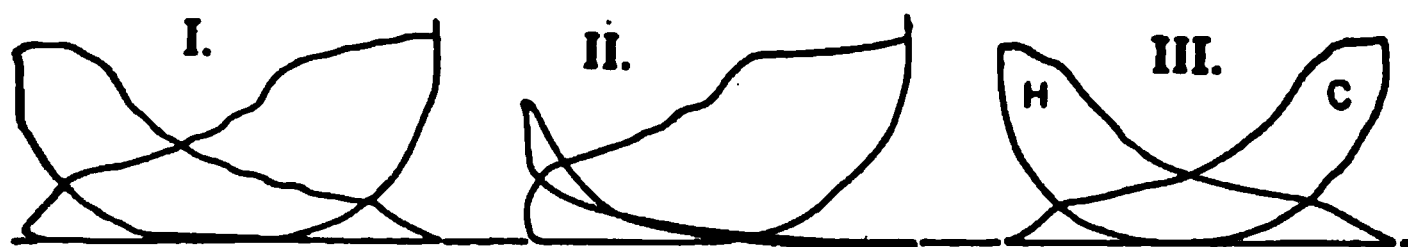


FIG. 347.—Valve-setting with the Indicator.

The matter of proper setting of the eccentric will be discussed in connection with the Corliss valve-gear. In any case, a thorough understanding of the working of the mechanism is fundamental to an intelligent treatment of faults in its operation.

#### § 48. The Shifting Eccentric: Variable Steam Distribution.

(a) MOVING THE ECCENTRIC-CENTER.—Following the line of development suggested by Fig. 338 III. and IV., and carried forward in Relation 3 under Fig. 345, we see that if the center of the eccentric be shifted along a line at right-angles to the crank-arm, changing both the length and the inclination of the eccentric-radius, the cut-off will be varied without changing the lead. This fundamental principle of the whole class of single-eccentric variable cut-off engines (as well as several allied forms) is illustrated in Fig. 348.

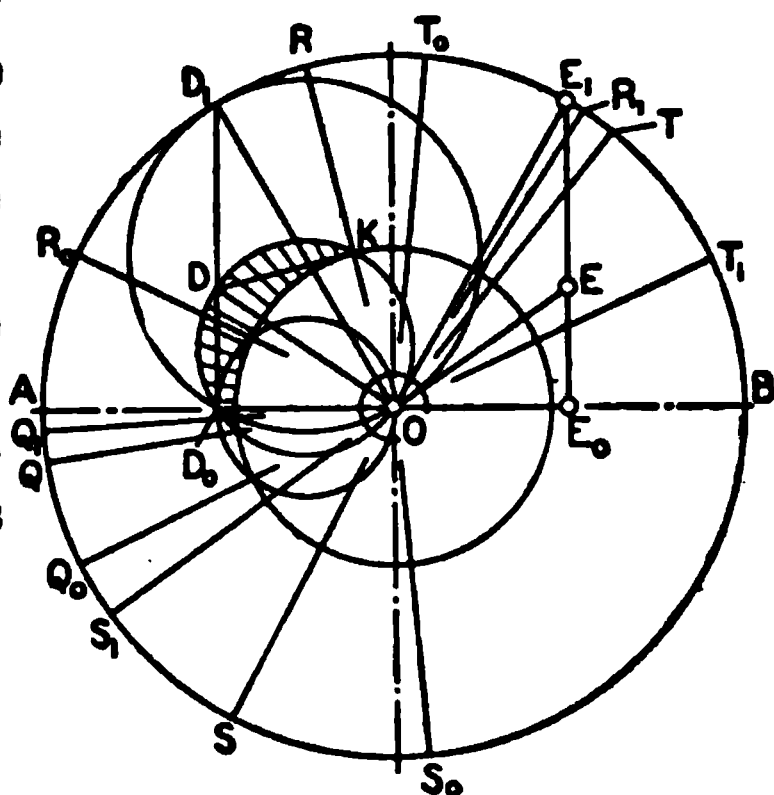


FIG. 348.—Shifting Eccentric with Constant Lead.

The eccentric is supposed to be carried on a cross-slide keyed to the shaft so that the center can be moved along the path  $E_1E_0$ , and is either clamped in any particular position or held in place by the governor. For the longest radius  $OE_1$  the valve



effly in that there is here an actual eccentric, surrounding the shaft, instead of a small pin off the end of the shaft. The whole piece PQ is called the swinging eccentric or the eccentric-pendulum: the center E moves along an arc struck from P, and its position is controlled by the governor, after the manner explained in § 2 (j).

The working of the governor is another subject, and will be taken up in the next chapter. For present purposes, all we need to know is the shape of the locus of E and the limits of its length. The required dimensions are shown on Fig. 350, where different possible arrangements are given and a conventional method is developed for stating, as concisely as possible, the essential data.

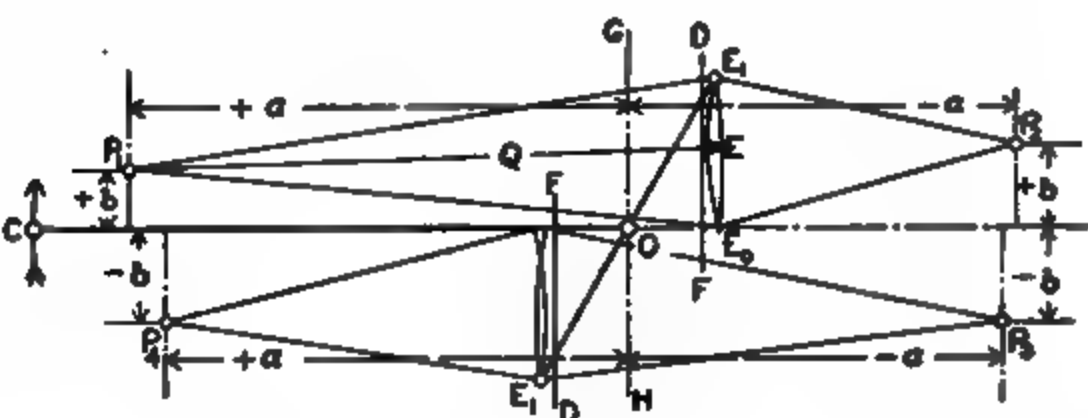


FIG. 350.—The Eccentric-pendulum

With the crank-line CO and the perpendicular GH as axes, the pivot P is located by the co-ordinates  $a$  and  $b$ , the algebraic signs having the meanings indicated. In general, for any position of the crank,  $a$  is plus from O toward C,  $b$  is plus in the direction of the motion-arrow through C. Then the pendulum-radius or PE, and the limiting eccentric-radius  $r_1$  or  $OE_1$ , complete the data. In the absence of specific statement,  $E_0$  is supposed to be on the line CO.

Of the four characteristic positions of P, numbers 1 and 2 belong to the direct, 3 and 4 to the indirect. The offset  $b$  may be made equal to  $r_1$ ; making it equal to half the radius is the nearest approximation to the best. In any case, drawing a line DF

at a distance equal to the steam-lap  $s$  from GH will show clearly how the lead will vary as E changes its position.

(c) DIAGRAMS FROM AN AUTOMATIC CUT-OFF ENGINE.—In the example worked out in Fig. 351, the data are:  $a = +5\frac{1}{2}"$ ,  $b = +1\frac{1}{2}"$ ,  $Q = 6\frac{1}{2}"$ ,  $r_1 = 2\frac{1}{2}"$ ,  $s = 1"$ ,  $i = 0$ . The eccentric-locus, from  $E_1$  to the crank-line, is divided into three equal parts, and four Reuleaux diagrams are drawn, a part of the lap-circle helping to locate the

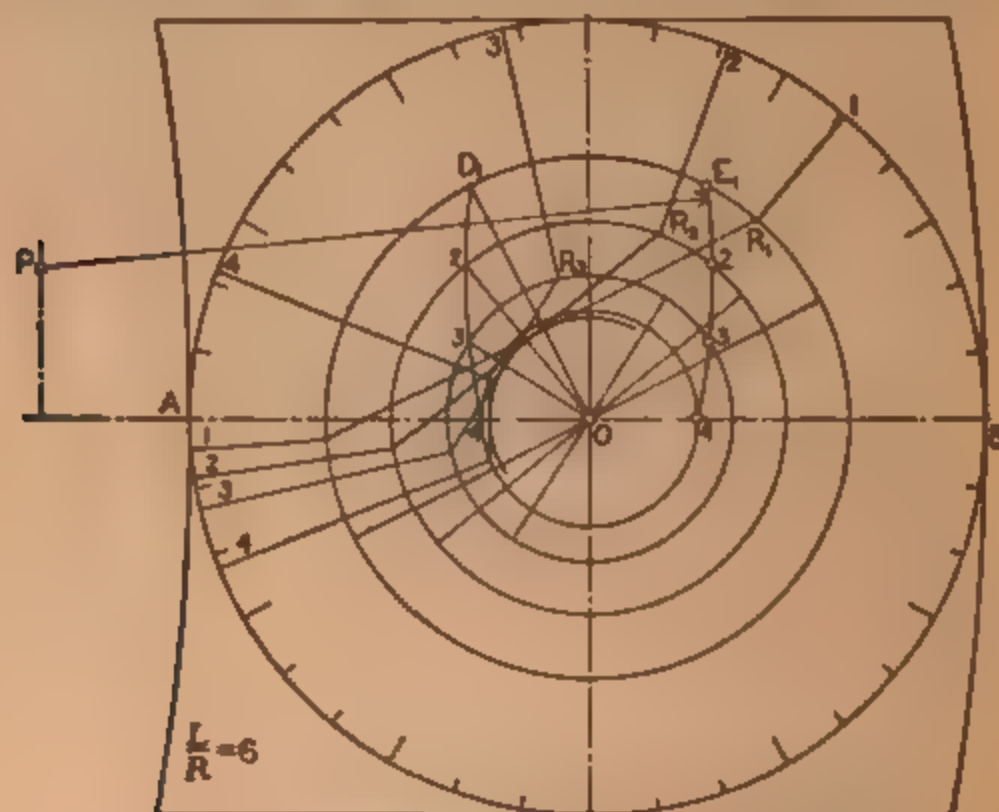


FIG. 351 -Valve-diagrams from a Shaft-governor.

several lap-lines. The change in admission and cut-off is shown by drawing in the corresponding crank-positions, while release and compression would be located by simply extending the base-lines of the diagrams.

The curves in Fig. 352 are plotted from the diagrams in Fig. 351. The four distances  $QE$ ,  $Q'E'$ ,  $NH$ ,  $MH'$ , are all equal to the port-width  $b$ , which is  $2"$ : and even with the full stroke of the valve the port is not fully opened for admission, though there is over-travel on the exhaust side in this one case. The cut-off is only a moderately early cut-off and yet we see the effect of port-opening, and how great an effect can be produced.

light change in the length of the valve-rod. Note that in case where the eccentric is in line with the crank, the motion-diagram reduces to a single line, which would be straight with the piston in harmonic motion, but is here slightly curved.

(d) PROBLEMS ON THE SHAFT-GOVERNOR.—The following data from actual engines, as were those for Fig. 351. In each problem, get the eccentric-locus first, and then draw valve-diagrams for the greatest eccentric-radius  $r_1$ , for cut-off at one-third of the stroke, and for the earliest cut-off. One-third is chosen because it will give an effective cut-off, referred to the boiler-pressure, about one-quarter of the stroke; and it is upon this cut-off that the rated power of the engine is based. The Zeuner diagram is rather better for illustration, the Reuleaux for accurate determination of the whole movement.

PROBLEM 8. Direct valve,  $a = +14''$ ,  $b = 0$ ,  $Q = 15''$ ,  $r_1 = 1\frac{1}{2}''$ ,  $s = \frac{1}{4}''$ ,  $i = 0$ .

PROBLEM 9. Indirect valve,  $a = -1\frac{1}{2}''$ ,  $b = -\frac{1}{2}''$ ,  $Q = 2\frac{1}{2}''$ ,  $r_1 = 1\frac{1}{2}''$ ,  $s = \frac{1}{4}''$ ,  $i = \frac{1}{4}''$ .

PROBLEM 10. Direct valve,  $a = +5\frac{1}{2}''$ ,  $b = +\frac{1}{2}''$ ,  $Q = 6\frac{1}{2}''$ ,  $r_1 = 1\frac{1}{2}''$ ,  $s = \frac{1}{4}''$ ,  $i = \frac{1}{4}''$ .

(e) WIDTH OF PORT-OPENING.—Ideal valve-action would be characterized by a very quick (practically "instantaneous") movement in opening and closing the port, together with a very small width of opening. Fig. 352 shows how far these requirements fail of fulfilment, especially when the cut-off is early. Let us suppose that in this particular engine the width of the port is proportioned so as to give a certain maximum velocity of steam-flow, say 200 ft. per sec., after the manner of § 43 (u). To know what width of opening would give this same velocity at any point in the stroke, we draw the dotted curves  $V_1$  and  $V_2$  in Fig. 352, using the factors in Table X., which are, according to § 35 (d), ratios of piston-velocity to crank-pin velocity as well as ratios of force. Equality between rate of steam-flow and rate of piston-displacement requires that, for a certain constant steam-velocity, the area or width of the port-opening shall vary as the velocity of the piston. We therefore multiply the total width of the port belonging to each crank-angle, and lay off the result

as an ordinate from the lap-line or port-edge, QR, Q'R'. Here the width of port is taken to correspond with the velocity at 90°, or that of the crank-pin, so that the curves go just a little way outside the line EF, E'F'. One very evident conclusion is that the actual velocity of flow past the valve must greatly exceed the assumed value of 200 ft. per sec., especially at early cut-off, with consequent large drop in pressure toward cut-off. Another point is, that the opening ought to be a little greater at head end than at crank end, on account of the somewhat higher velocity of the piston in the first part of the forward stroke.

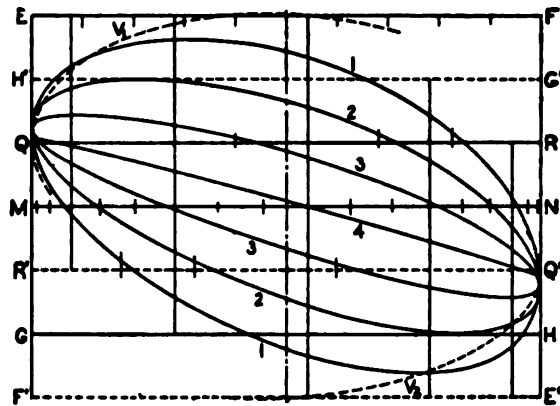


FIG. 352.—Stroke-line Diagrams from Fig. 351.

(f) SYMMETRICAL ADMISSION.—The possibility of equalizing the cut-offs—in the sense of making them take place at the same fraction of both strokes—by changing the laps, has been already suggested under Fig. 336. In Fig. 352, the mean cut-off for each curve (or that which would occur with harmonic motion of the piston) is indicated by the short cross-lines; and it appears that the amount of change in lap required varies with the driving eccentric. These curves are re-drawn in Fig. 353, with the laps changed so as to give equality for curve no. 3, which will now show cut-off at four-tenths of each stroke.

This figure illustrates excellently the important principle that equality in the amounts of steam admitted—a far more essential

requirement than mere symmetry in valve-events—depends not only upon the length of the period of admission, but also upon the width of opening. The throttling effect would now be so much greater in the head end than in the crank end that the steam diagram would be much smaller in area.

With small openings, the diagram will be very sensitive to slight changes; so that, in adjustment under the indicator, it

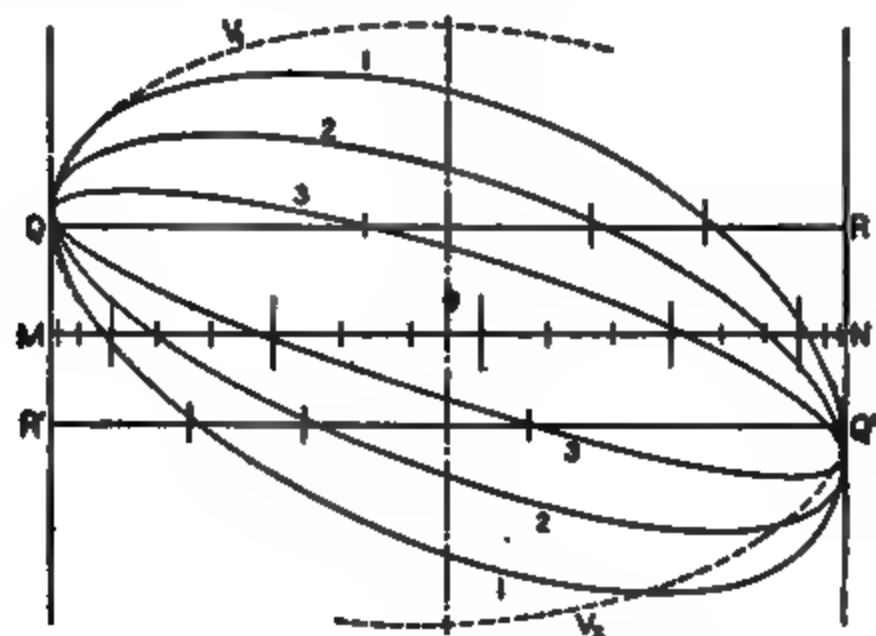


FIG. 353.—Cut-offs Equalized.

could be found that only a minute alteration from the conditions in Fig. 352 would be needed to make the admissions equal for the smaller powers of the engine.

(g) INDICATOR DIAGRAMS.—To illustrate the matters just discussed, and also to show the characteristic action of a valve-gear with shifting eccentric, the set of diagrams in Fig. 354 is derived from Fig. 352. These are made-up diagrams, not exactly determinable from the valve-action, but also based on experience with engines of this class. Referring to § 18 (d), we see that to know the point on the stroke at which the valve closes is, in itself alone, a very insufficient determination of the amount of steam admitted. With this we must know, or be able to estimate fairly well, the drop in pressure due to throttling, down to the mechanical cut-off. Thus the cut-off ordinate at G can be located by simple projection from Fig. 352: but it is only from experience that we





which were taken from a locomotive on the same run and with the same valve-action—that is, with the reverse-lever in the same notch—but at different speeds. These were, 30 miles per hour for the larger, full-line diagram and nearly 80 for the smaller, equivalent to 128 and 330 R.P.M. respectively, with 80-inch driving-wheels. The diagrams are reproduced in true proportion, except that the difference between the back-pressure lines is slightly exaggerated. The throttle-valve was a little wider open

for II. than for I., but even then the drop in pressure from the boiler (B.P.) to the steam-chest (S.C.P.) was much greater at the higher speed: and there is a similar increase in the loss of pressure by the steam in getting into the cylinder. The indi-

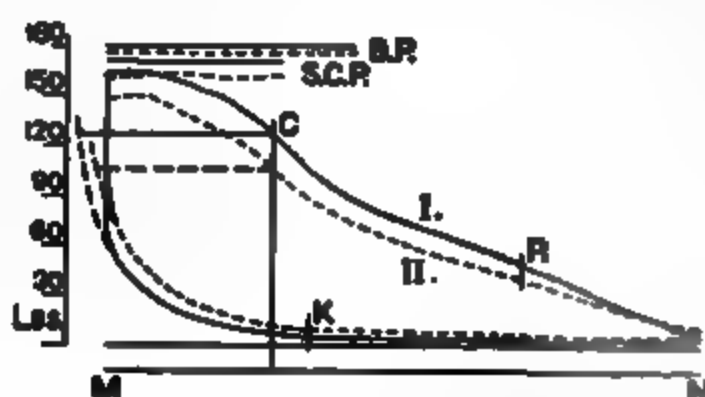


FIG. 355.—Diagrams at Different Sp

cated steam consumption is just about the same, however, for the two diagrams; and since the cylinder-condensation is less at the higher speed, the smaller diagram represents a higher thermodynamic efficiency, in spite of the increased wire-drawing effects.

(h) VALVE-GEAR PERFORMANCE.—This is shown in ultimate terms by curves such as are drawn in Fig. 356. The base-line represents the eccentric-locus developed; which is chosen, out of several possible bases, partly because it puts the data from the valve-gear into good shape for use in a discussion of the action of the governor. Curve I. shows the cut-off by the valve, EG/MN in Fig. 354, while II. is the effective apparent cut-off at boiler-pressure, EF/MN, got by producing the expansion-curve backward. It is noteworthy that the absolute difference between these curves is nearly constant in this case. Curve III. shows the length of the compression-period, likewise expressed as a fraction of the stroke: and IV. represents the manner of variation of the M.E.P., here given as a fraction of the gage-pressure, or of the boiler-pressure above the atmosphere in Fig. 354.

A set of curves like this, plotted from the actual indicator

cards, gives a very convenient and useful record of the performance of an engine. It is characteristic of the type of valve-gear that

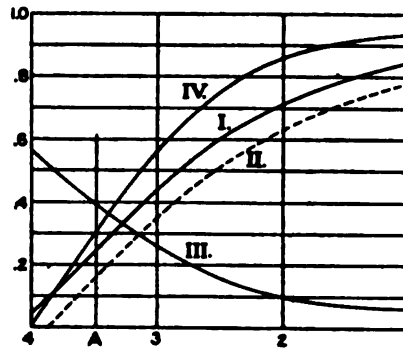


FIG. 356.—Valve-gear Curves.

the outer part of the eccentric-locus, from 1 to 2, is of little use for governing the engine, because the M.E.P. varies so slowly—here only from about 93 per cent. to 86 per cent. of the boiler-pressure for this whole range of movement.

By plotting results from the valve-diagrams of a number of engines of the shaft-governor class, the writer has found an average law for the relation between compression and cut-off to be

$$n = 0.5 - 0.8m + 0.4m^2, \dots \dots \dots (312)$$

where  $m$  is the apparent cut-off by the valve, the same as in Eq. (89), § 18, and  $n$  is a similar measure of the period of compression, as in curve III. on Fig. 356.

#### § 49. Secondary Influences and Special Cases.

(a) EFFECT OF THE ECCENTRIC-ROD.—While the distortion from harmonic motion, on account of the angular movement of the eccentric-rod, is usually very small, it must sometimes be taken into account. In the fundamental diagram in terms of the eccentric-angle (the primary form of Fig. 332), this is done by striking an arc with the rod-length as radius, after the manner of Fig. 111. To show the travel of the valve from its true mid-position, we should pass this arc through the center  $O$  as in Fig. 357 I.: and it is very easy to see that the travel-distance  $ES$  will be almost exactly as given by the harmonic diagram when the valve is near the ends of its stroke, but that the error in this simple diagram will increase as the valve approaches mid-stroke.

The "true mid-position" just referred to is midway between the extremes of valve-movement; or, as a more general description, it is located by measuring off from  $O$ , upon the stroke-line, the rod-length  $l$ . But while this is, at first sight, the obvious datum-point from which to estimate valve-travel, it is not the point with reference to which the travel is most closely given by the common valve-diagrams, as will now be made clear.

In Fig. 357 II., the crank-eccentric is placed on the two dead-centers, at  $C_1OE_1$ ,  $C_2OE_2$ ; and arcs with  $l$  as radius are drawn at the same distance (equal to the lead) inside of  $E_1$  and  $E_2$ . Now it is entirely reasonable to assume that equal laps are correlative

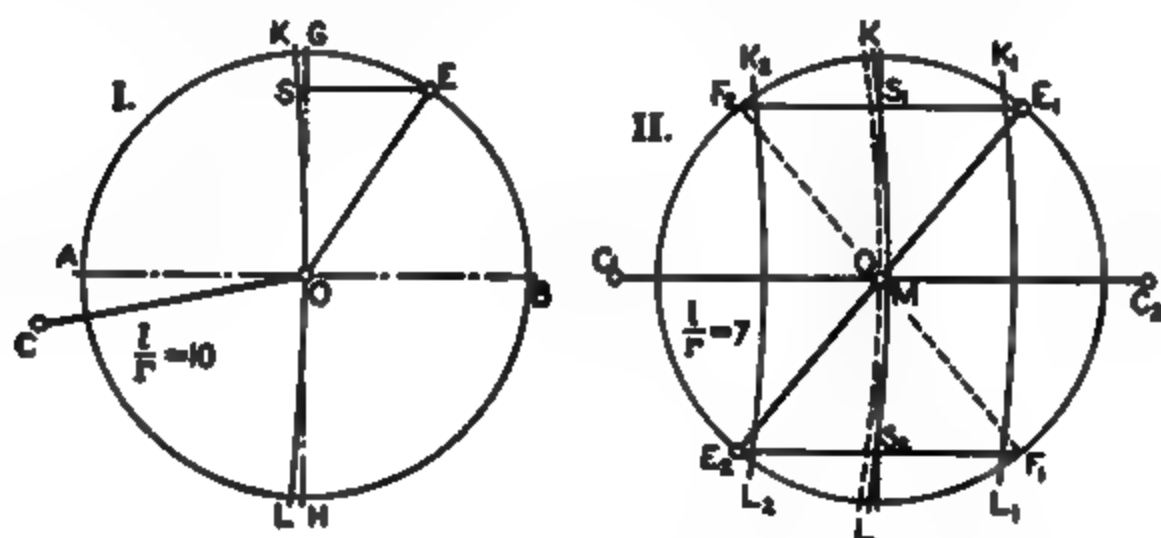


FIG. 357.—The Center of Movement.

with equal leads; and if this is true, then the distances from  $E_1$  and  $E_2$  to the reference-arc for mid-position must be equal, or this curve must agree with GH at the points  $S_1$  and  $S_2$ , rather than at the center  $O$ . Striking off  $l$  from  $S_1$  or  $S_2$ , we should locate the corresponding reference-point at the distance  $OM$  from the middle of the valve-stroke. Using this as the center of movement, the valve-diagrams will give the travel very exactly at admission and cut-off, where there is most need of its being accurately determined; and will have an error, here much exaggerated by the use of short rods, for very long and very short travels. The rod-ratio  $l/r$  lies usually between 15 and 25, with 20 as a good average value.

An interesting fact brought out by Fig. 357 II. is that, with

equal leads, port-openings due to right-hand movement of the eccentric are less than those due to movement toward the left. It is desirable that the openings be larger, if anything, for the head end than for the crank end, because the piston moves faster in the first part of the forward stroke than of the return stroke. With the common, direct valve, then, the effect of angular movement of the eccentric-rod is against equality in the steam-distribution: and an arrangement in which movement of the eccentric to the left serves the head end—as with an indirect valve or with a direct valve indirectly driven—has a small inherent advantage over the opposite arrangement.

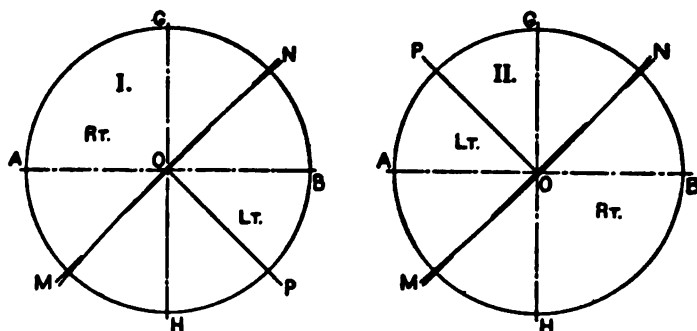


FIG. 358.—Rod-effect in the Reuleaux Diagram.

(b) THE EXACT REULEAUX DIAGRAM.—Rotating Fig. 357 II. backward through the angle  $\delta$ , we get exact Reuleaux diagrams as shown in Fig. 358, I. for the direct valve, II. for the indirect. For the latter, with this diagram, it is better to reverse the direction-meaning of the ordinate, since left-hand travel, along OP, is what goes with opening of the left or head-end port. This is simpler than to produce each crank-line through the center and take the ordinate from the point where it hits the other side of the circle (at the same time reversing the base-curve). But with the Zeuner diagram it is better to adhere to the usual conventions, and draw a single valve-circle, for the indirect valve, in the third quadrant, as in Fig. 342 II.

In any case, under the usual rule for locating OP, the direction

convexity of the reference-curve MN is determined wholly by eccentric-setting, without regard to the manner of driving valve. A reversing rocker-arm changes the direction-meaning the ordinates in Fig. 358, without affecting the curve; and, hinted at the end of the last paragraph, interchanges between ends of the cylinder the effects of the eccentric-rod in modifying the steam-distribution.

The Reuleaux diagram has an advantage over the Zeuner in that this action can be shown directly: an exact polar diagram can be got only by plotting successive ordinates and tracing a special curve.

(c) ROD-EFFECT WITH SHIFTING ECCENTRIC.—This is illustrated in Fig. 359, where the four eccentric-circles from Fig. 351 are reproduced, with the locus of E at C and at DF, in the two positions for crank on dead-center. The two reference-arcs  $K_1L_1$ ,  $K_2L_2$ , are drawn, after the manner of Fig. 357 II., so as to give equal leads for latest cut-off: and it is clear that the leads will become increasingly unequal as E moves in along EC and out along DF. This effect is here a good deal exaggerated; but a suggested alteration would be to equalize the leads for the early cut-off which mostly prevails, and let the extreme conditions be non-symmetrical.

(d) INFLUENCE OF THE LEAD.—The matter of proportioning this element of the valve-action so as to get the best results is one that can hardly be reduced to definite terms. In general, two objects are to be kept in view: first, to have the engine run smoothly, without a too sudden reversal of pressure at the end of the stroke; second, not to waste any of the possible area of the steam-diagram by late admission. Where the engine has what we may call a rotary load, or *delivered the same power* at a slight variation

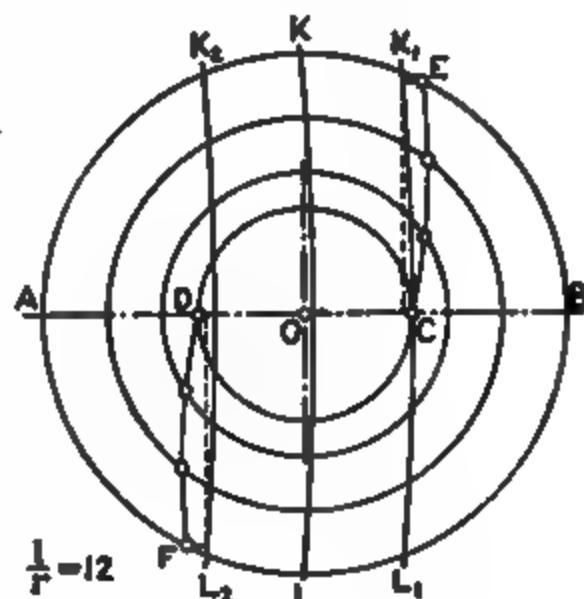


FIG. 359.—Rod-effect with Shaft-governor.

in the time when the valve opens can have little effect: but where the load acts directly upon the piston-rod, as in a pump with fly-wheels especially, the growth of the driving-pressure should barely precede that of the resistance, at the beginning of the stroke; and this may call for a very small or even for a negative lead. As to the matter of getting the full steam-pressure before the piston begins its stroke, it is evident that speed, clearance-volume, and height of compression all enter as important elements. In any case, the angle of lead—a measure of time—is rather more important than the absolute width of opening. In engines where the admission is controlled by a fixed eccentric, this angle will usually lie between  $5^{\circ}$  and  $10^{\circ}$ .

(e) VARIATION IN LEAD.—Engines of the shifting-eccentric type, with the shaft-governor, show considerable variety in the location of the pivot-point P, Fig. 350, and in the manner of variation of the lead. As typical cases in this respect, consider the diagrams sketched in Fig. 360. The point P is on the same side of

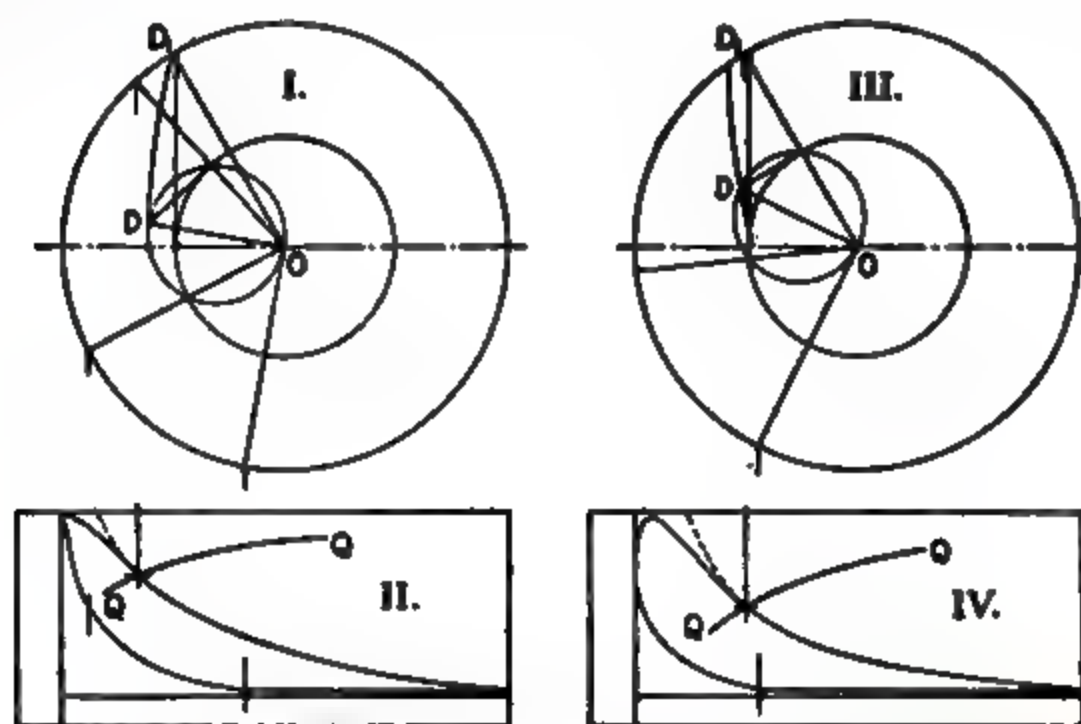


FIG. 360. — Different Ways of Varying the Lead.

the center in both I. and III; but in the first case it lies on the crank-line, in the second it is on a line through  $D_1$ . The valve-diagram in I. is drawn for cut-off at one-sixth of the stroke; and through

the large lead quite a wide opening of the port is secured, even with this very early cut-off. In III. and IV., the mechanical cut-off, or instant of complete closure, is at one-quarter stroke: the greater drop in pressure, due to throttling on account of the small opening, is clearly shown, especially by the lower position of the curve QQ.

One advantage of the arrangement at III. is that the governor can completely shut off steam from the engine, which it cannot do with the proportions in I. Further, the eccentric moves through a somewhat smaller distance, for a given range of power, in III. than in I. It is suggested by the steam diagrams, II. and IV., that the question as to the best method of governing the engine under small loads is here involved—this question being, whether it is better to throttle the steam or to make the cut-off very early. Without more than touching the outskirts of this rather extensive subject, we may remark that when cylinder-condensation is taken into account, a small-power diagram like IV. may give just as good economy as one like II., in spite of the increased throttling effect.

(f) EFFECT OF LOST MOTION.—In a mechanism like the valve-gear, with a number of joints—and especially in the complicated types, such as the link-motions and the Corliss gear—there may be considerable lost motion between the eccentric and the valve, if the joints are allowed to become at all loose. The effect upon valve-movement and steam-distribution can be very nicely shown upon the diagram, as is illustrated in Fig. 361.

It is safe to assume that friction of the valve is the predominating force in the valve-gear—even though, at high speeds and with a well-balanced valve, inertia-force may have a greater value near the ends of the valve-stroke. Under the control of friction, the valve will lag behind its ideal position by an amount equal to half the total free-play in all the bearings. In I., this is shown by shifting the semicircle GMD to the left, and FNE to the right, striking them from the centers  $O_1$  and  $O_2$ , respectively. That is, each half of the motion-circle is given a displacement opposite to the direction of valve-motion in the stroke which it represents. The directions “right” and “left” refer, of course, to movement of the valve as represented by the diagram, so that GF is right,

FG is left. In ideal working, each journal will be concentric with its bearing; when loose, half of its total free motion will bring it into contact on either side. At each end of the stroke the valve will pause, as shown by the straight lines DH and EK, while the "back-lash" in the mechanism reverses.

Now all the events of the steam-distribution will be a little late. In the right-hand stroke these are, taking them in order: crank-end cut-off R', head-end compression S, crank-end release T', head-end admission Q. And it is evident that this genera

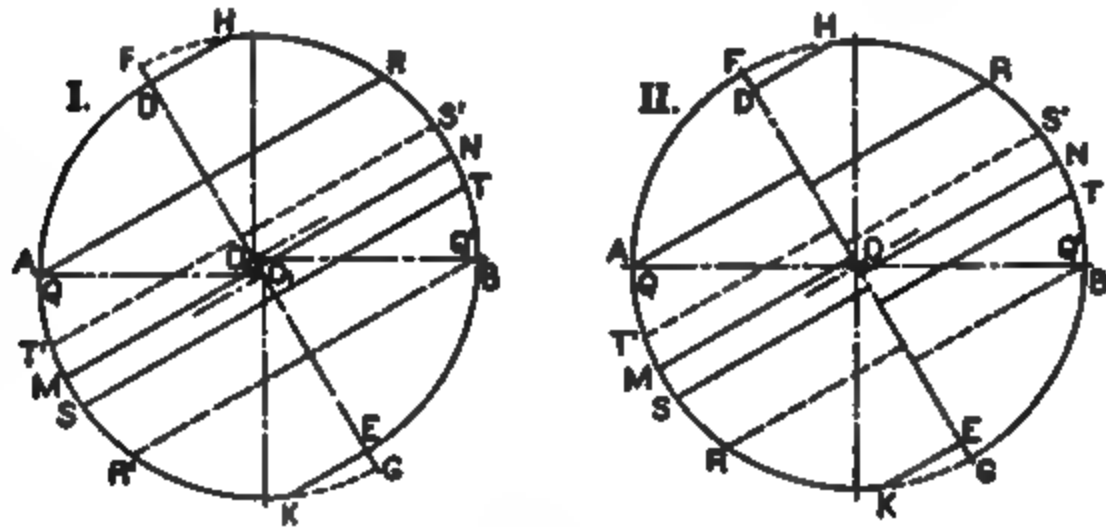


FIG. 361.—Effect of Lost Motion.

retardation could be compensated by giving the eccentric a slight extra advance.

Instead of displacing the two halves of the circle, it is rather simpler, graphically, to shift the reference-lines in the other direction, as at II. Since these can be thought of as representing the valve-seat, we move them ahead, in the direction of the movement of the valve, so that in the ideal movement represented by the circle the valve would take a little longer to reach any particular position.

(g) EFFECT OF A NON-CENTRAL STROKE-LINE.—The general case of the slider-crank mechanism, in which the stroke-line of the slide does not pass through the center of the shaft, finds frequent application in one class of valve-gears, the "link-motions": and is sometimes used in single-eccentric gears, for a purpose which is discussed in the next article. To see how the movement of the slide is modified from that in the usual case, consider Fig. 362.



First of all, the rod-length  $l$  is measured off from  $O$  to  $P_0$  and swung about  $O$  so as to locate on the stroke-line  $MN$  the ideal mid-position  $P$ . Then from  $P$  is struck the arc  $KL$ ; and for any position of the eccentric (or crank), as  $OF$ , we have only to make  $OQ = l$ , draw  $FT$  parallel to  $MN$ , and join  $PT$ , in order to get a parallelogram  $FTPQ$  and show that  $FT$  is always equal to the travel-distance from  $P$ .

Now the dead-points are not at  $A$  and  $B$ , but at  $C_1$  and  $C_2$ , where the crank and rod coincide on a line through the center: and the total travel is greater than the diameter of the circle, as appears when we locate  $M$  and  $N$  by striking off the lengths  $AP_0$  and  $BP_0$  from  $O$ . The reason for this excess is most clearly seen by drawing the distance-lines  $C_1D_1$ ,  $C_2D_2$ : which shows also that the two ex-

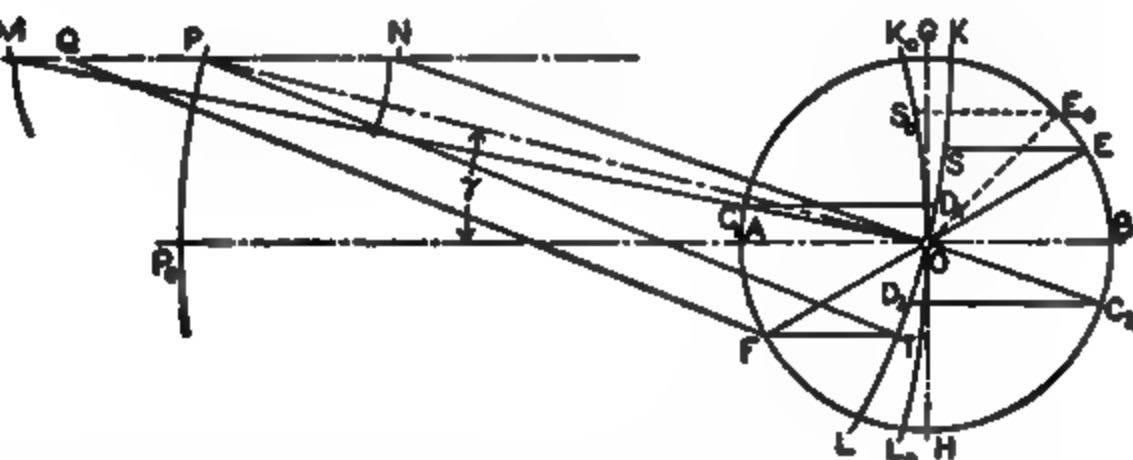


FIG. 362.—Non-central Stroke-line.

same distances will be unequal, partly because  $C_2$  is farther from  $B$  than is  $C_1$ , partly on account of the curvature of the reference-line  $KL$ .

For a very long rod, the curve  $KL$  would be replaced by a straight line at right angles to  $OP$ , which latter is the line of mean rod-slant.

Approximately—the approximation being less close as the angle  $\gamma$  is greater and the rod is shorter— $Q$  travels as if it were driven along  $P_0$  by the eccentric  $OE_0$ , located from the actual eccentric  $OE$  by making the angle  $EOE_0$  equal to  $\gamma$ : that is, the change in the mean direction of the rod from  $OP_0$  to  $OP$  (equivalent to a change in the eccentric-angle) has the greatest influence upon the travel

of the slide; and the other effects, such as that due to the angle between  $OP$  and  $MN$ , are secondary.

(h) **EQUALIZING THE CUT-OFFS.**—By combining an offset stroke-line with the use of a very short rod, the desideratum of unequal (angular) periods of port-opening in the two ends, with symmetrical beginnings, can be secured: which makes possible equal cut-offs with equal laps. Thus in Fig. 363 the valve is supposed to open at the diametrically opposite eccentric-positions  $OR$  and  $OS$ . The arcs  $RS$  and  $TU$  are drawn like  $K_1L_1$ ,  $K_2L_2$ , on Fig. 357, being struck with  $l$  as radius from the points  $G$  and  $H$  on  $MN$ . On account of the curvature of these port-edge lines, the angle  $ROS$  is greater than  $TOU$ —in this case about  $136^\circ$  as against  $127^\circ$ , according to the protractor-scales on the figure.

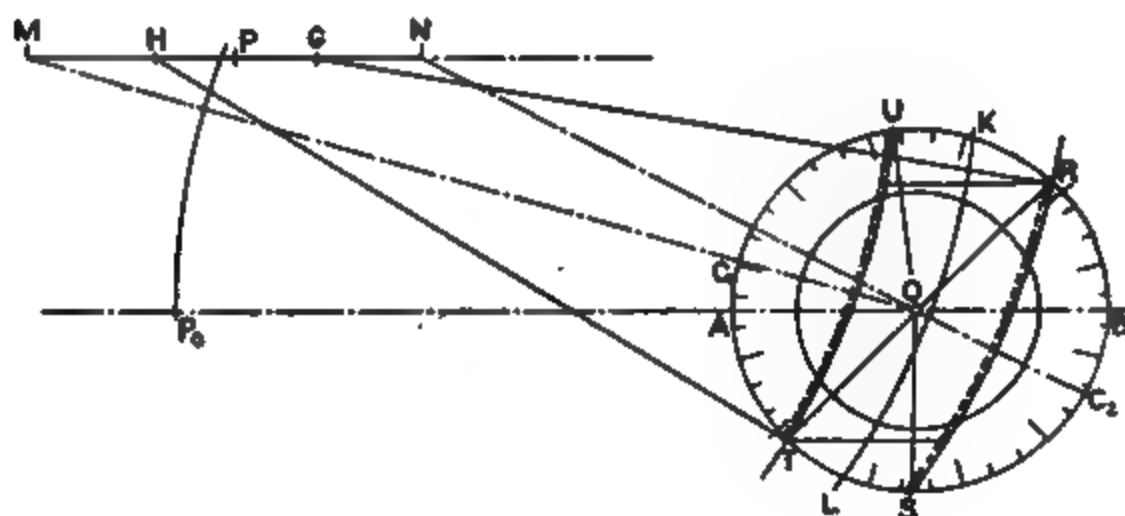


FIG. 363.—Unequal Periods of Opening.

With this very short rod, the non-symmetry of the widths of opening is quite marked: and if the valve be shifted just a little on its seat, as by lengthening the valve-rod, so as to bring the lap-lines to the dotted positions, then the difference between the angles of closure will be increased. It will be noted that the change in leads here involved is just the opposite of that in Fig. 353.

An application of this device will be found in the valve-gear discussed in § 58, under Figs. 495 to 502. That it could not be used with a shifting eccentric is shown by drawing the small inner circle, and noting the great distortion-effect analogous to that in Fig. 359: and here the short rod is essential, for it is only

because of the curvature of RS and UT that the arcs UR and T are unequal.

### § 50. The Stephenson Link-motion.

(a) REVERSING THE ENGINE.—The derivation of this most common type of reversing valve-gear is at once suggested by consideration of the eccentric-settings for opposite directions of rotation, shown in Fig. 364. If the two eccentrics are put on

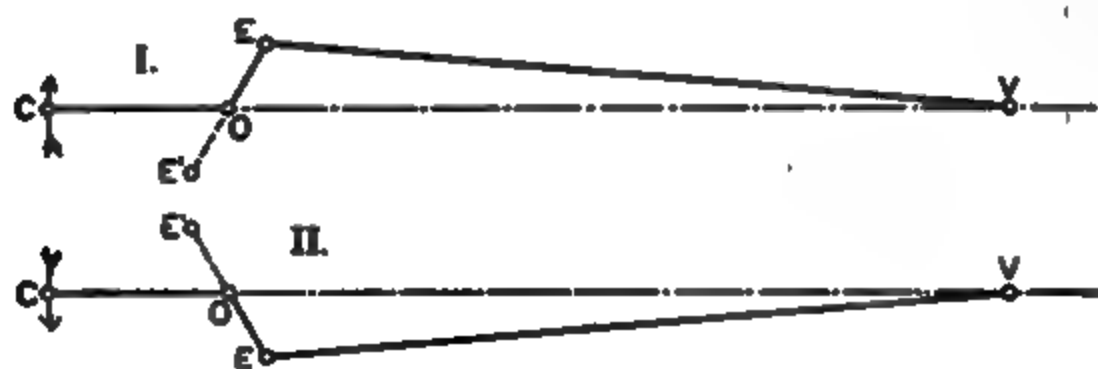


FIG. 364.—Eccentric-settings for both Directions.

the engine-shaft side by side, and if the control of the valve can be transferred from one to the other at will, the engine can be made to run in either direction. Logically the primary type of device for making this transfer is that shown in Fig. 365, where

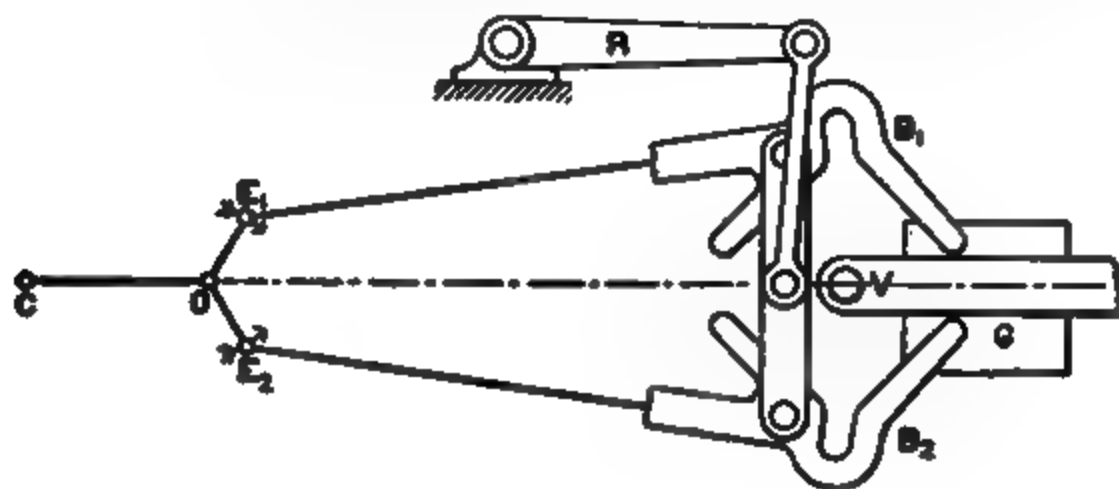


FIG. 365.—The Gab-motion.

by moving the lever R either hook,  $B_1$  or  $B_2$ , can be made to engage the pin at V. This crude arrangement preceded the link-motion, and was used on hoisting-engines: it has, of course, been entirely superseded.

(b) **THE STEPHENSON'S GEAR.**—The substitution of the curved link for the "rocker" of Fig. 365 is a radical improvement, not on account of its mechanical superiority, but because it ad

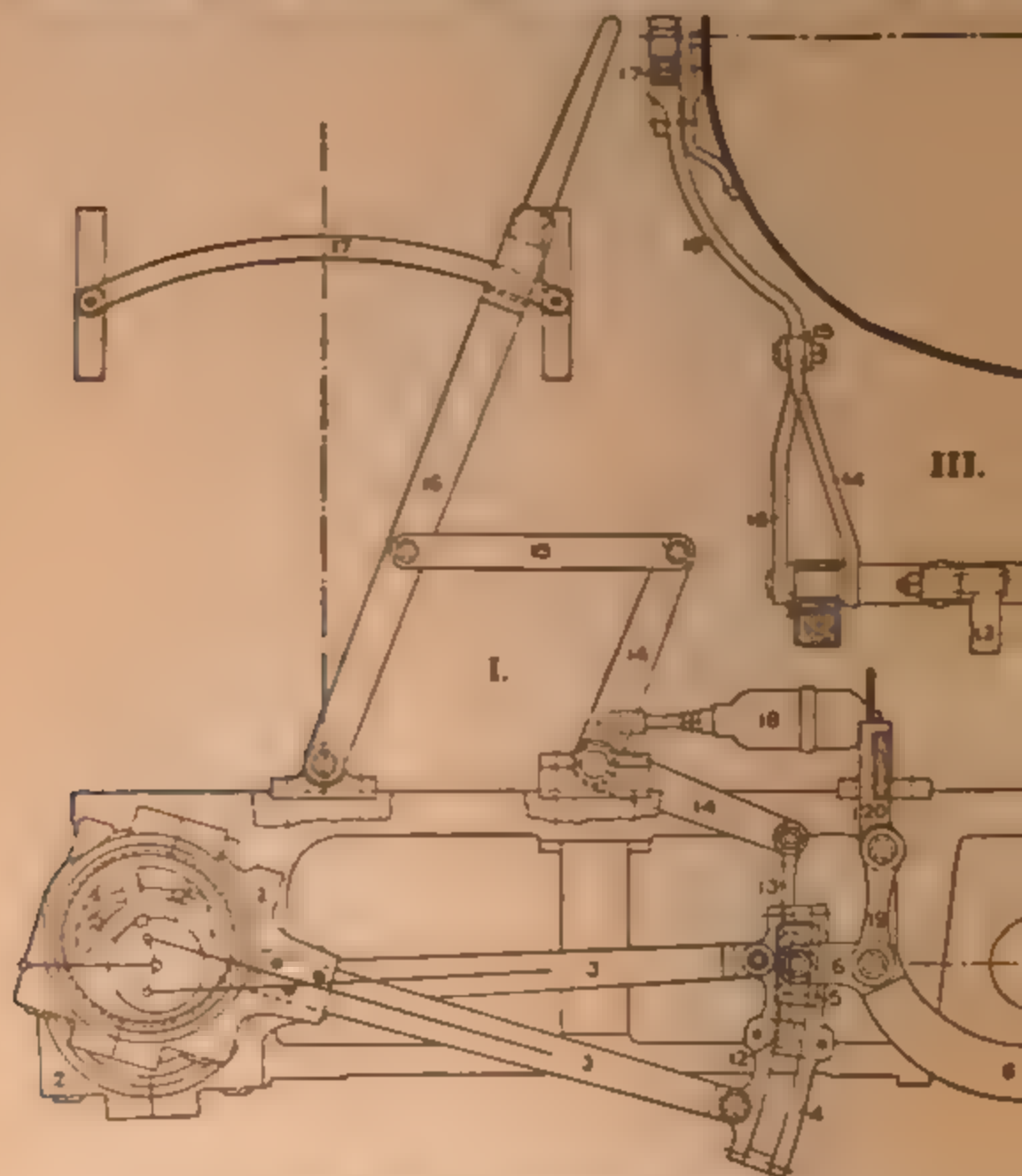


FIG 366.—A Locomotive Valve-gear.

second very important function to that of merely reversing the engine, in making possible a regulation and variation of the steam distribution very similar to that effected by the movable eccentric of the shaft-governor

Examples typical of the two most important applications of the link-motion—in the locomotive and in the marine engine

given in Figs. 366 and 368. In the first, the apparatus is complicated by reason of the extensive transmission-gear between the link and the valve. The several views are disposed with regard to the space available than to their strict relations in the scheme of projection. I. and II. make up the prin-

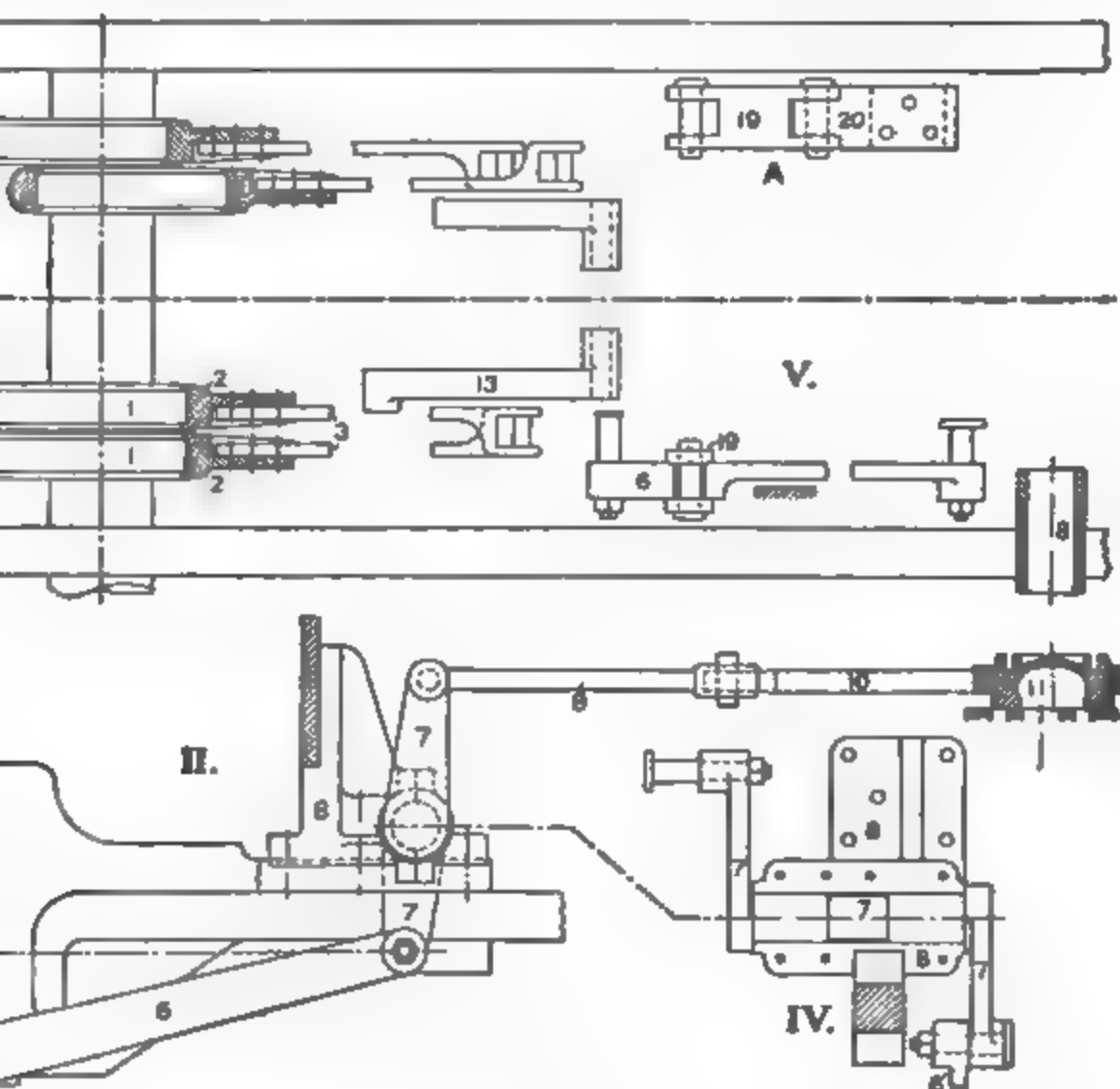


FIG. 366.—Continued.

- |                         |                           |
|-------------------------|---------------------------|
| 1. Eccentric.           | 11. Valve.                |
| 2. Eccentric-strap.     | 12. Hanger-pin.           |
| 3. Eccentric-rod.       | 13. Suspension-rod.       |
| 4. Link.                | 14. Lifting-shaft.        |
| 5. Block.               | 15. Reach rod.            |
| 6. Extension-rod.       | 16. Reverse-lever.        |
| 7. Rocker.              | 17. Notched Arc.          |
| 8. Rocker-box.          | 18. Balance-spring.       |
| 9. Valve-rod Extension. | 19. Extension-rod Hanger. |
| 10. Valve-rod.          | 20. Bearing for Hanger.   |

l or side view; III. shows the controlling or adjusting gear, in the front; IV. is a detail view of the rocker, and in V. a

number of the parts are laid out in plan, chiefly for the purpose of showing their relative positions as measured from a vertical

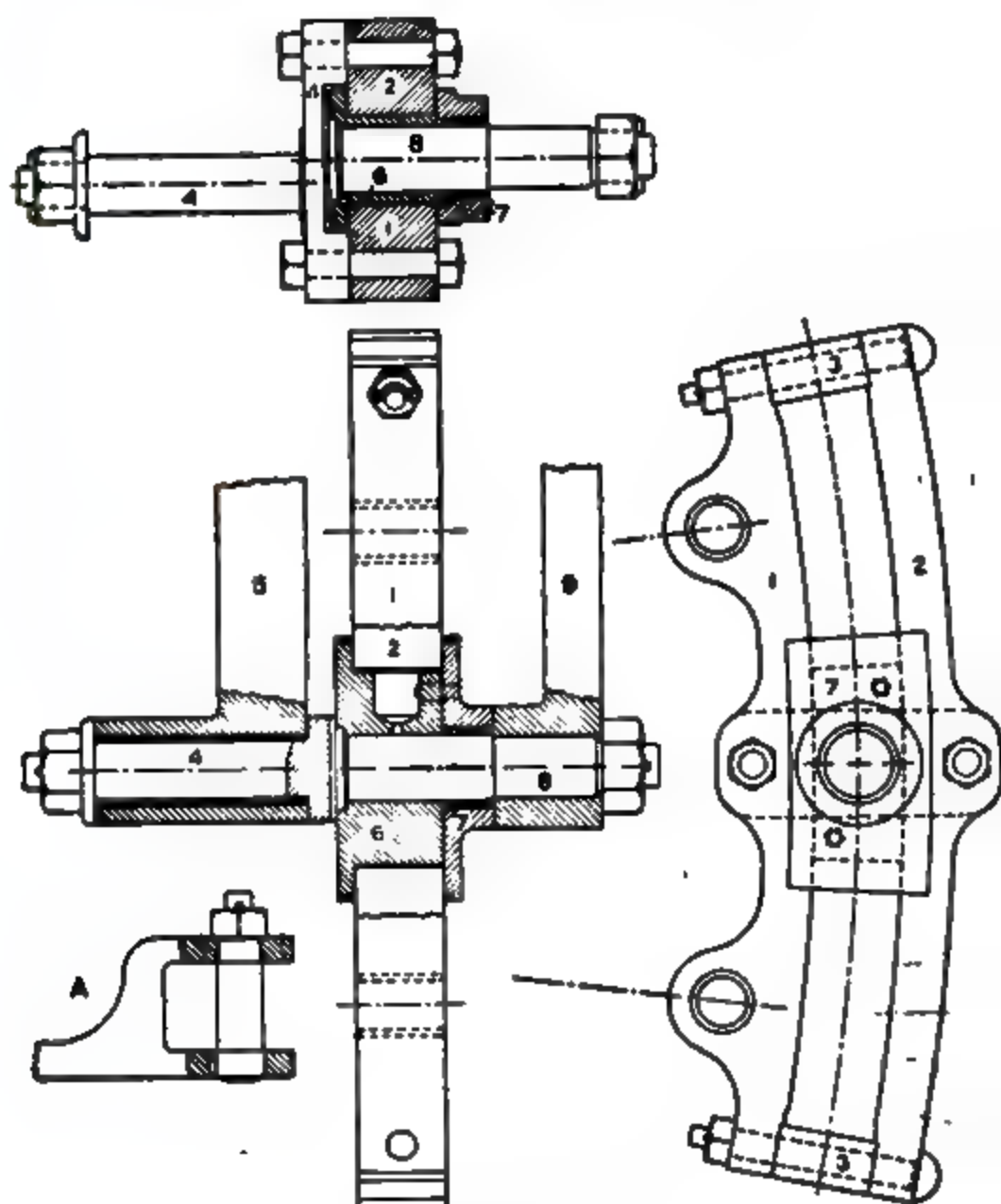


FIG. 367.—The Locomotive Link.

- |                           |                           |
|---------------------------|---------------------------|
| 1. Back Half of Link.     | 6. Block.                 |
| 2. Front Half of Link.    | 7. Loose Flange of Block. |
| 3. Distance-blocks.       | 8. Rocker-pin.            |
| 4. Hanger-pin and Saddle. | 9. Rocker-arm.            |
| 5. Suspension-rod.        |                           |

center-plane, and how near the mechanism comes to having members all in one plane.

The valve-gear as a whole divides itself into several component parts. The link-motion proper, from the shaft or

to the link 4, including the suspension-rod 13, forms one complete mechanism, with definite motion; the adjusting-gear, pieces 14 to 16, constitutes another division; the valve-connections, comprising everything from the block 5 to the valve, make up a third. The link, shown in detail in Fig. 367, is of the offset type: that is, the pin-connections for the eccentric-rods are set well to one side of the center-line of the link: whereas in Fig. 368 these are on the curved center-line. The radius of the link-arc is equal to the distance from its center-line to the eccentric-center, making the eccentric-rod: in other words, if the centers of both the eccentric-straps could be brought to the center of the shaft, then the center of curvature of the link would also coincide with the shaft-center. The construction of the link and block, and the arrangement of the suspension-pin, are so clearly shown in Fig. 367 as to need no explanation: but in one particular this arrangement differs from Fig. 366, namely, in that here the block moves the lower end of the rocker directly, instead of through an extension-rod.

The reverse-lever system, or the adjusting-gear, controls the power of the engine, by raising and lowering the link so as to change the position of the block in the link, and thereby vary the steam-distribution. By means of a latch on the reverse-lever, engaging notches on the arc 17, the valve-gear can be locked at any setting. In the casing numbered 18 is a compression-spring, which balances the weight of the link-motion, so that the reverse-lever will move easily in either direction.

The general form and the dimensions of the transmission-gear, from the block to the valve, are determined by conditions outside the valve-mechanism; that is, by the location of the main driving-axle and of the valve-chest. In locomotives of European construction the valve is commonly so placed that it can be directly driven.

The valve-gear of a marine engine is usually of this simpler, direct-driving type: and where, as in Fig. 368, the rod-pins are on the center-line of the link, some small disturbing elements are eliminated from the movement of the valve. The detail of the two-bar link at II. shows how the pins are got into this posi-

tion without interfering with the full movement of the block: while the alignment of the eccentrics with the link and the forked end of the eccentric-rod are given at III. The reversing-shaft

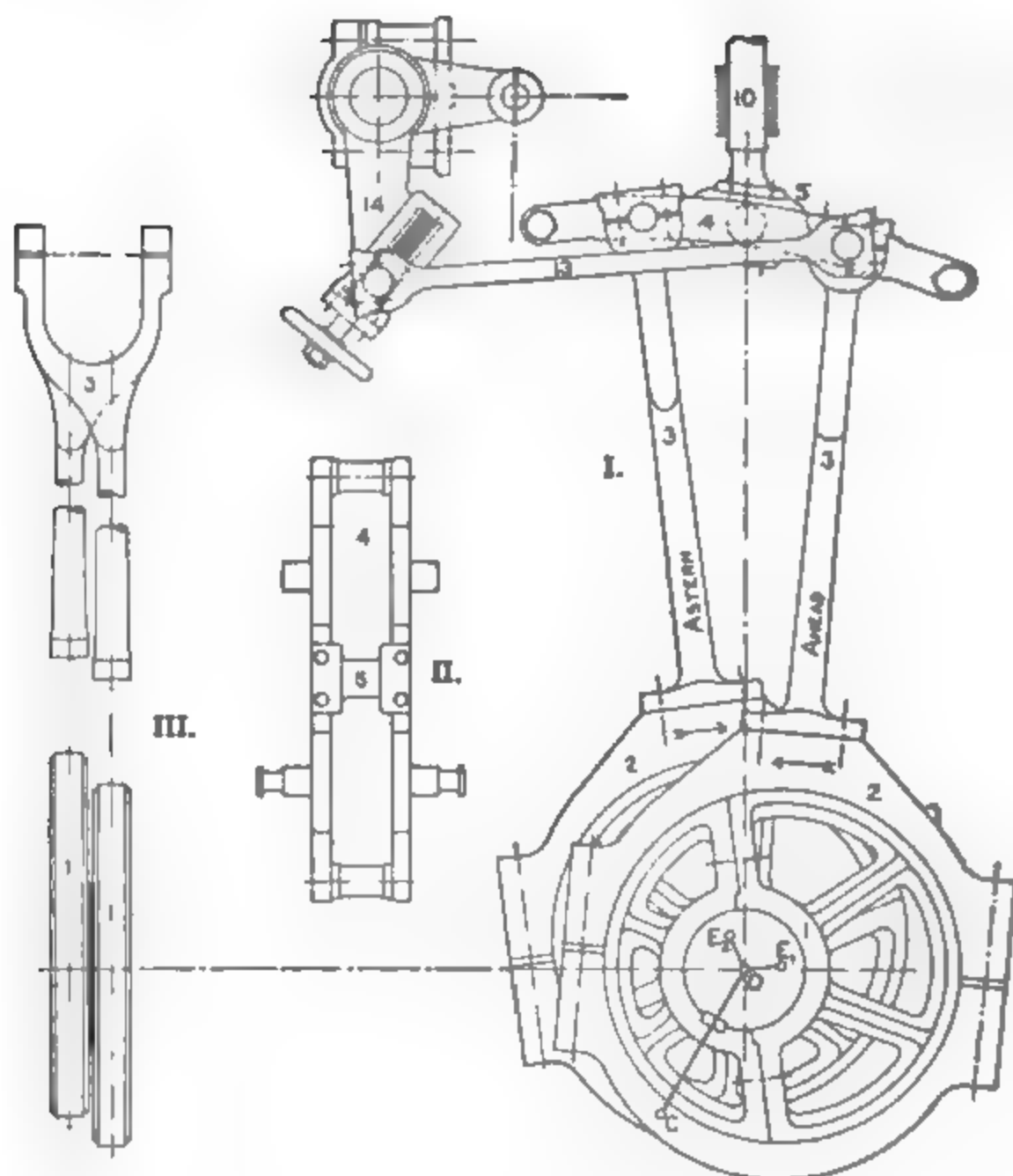


FIG. 368.—Marine Type of Link-motion.

Corresponding parts have same numbers as on Fig. 366.

runs along the front of the engine, and controls all the individual valve-gears together. By means of the adjustment device, the location of the suspension-pin on the end of the reverse-arm 14 can be changed, so as to vary the relative actions of the several



ves in the compound engine. The slot in which moves the  
 ck carrying the pin is at such an angle that adjustment will  
 re its full effect upon forward running, and practically none  
 on backward running of the engine. This particular gear  
 ongs to the low-pressure cylinder of the engine illustrated  
 Figs. 243, 259, and 260; and the eccentrics are made so much  
 ger in diameter than is apparently necessary because the other  
 o pairs are mounted on the coupling-flanges of the shaft-sections,  
 l all are alike in outer diameter.

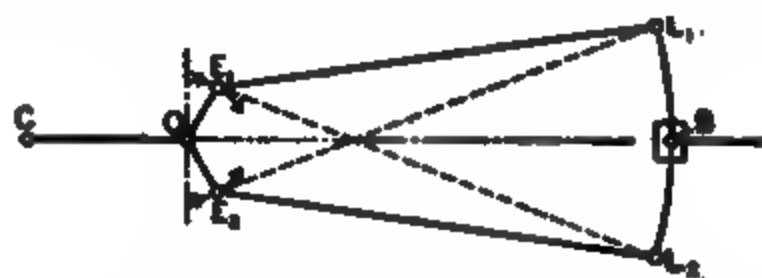


FIG. 369.—The Direct Link-motion.

(c) ARRANGEMENT OF THE RODS.—In Figs. 369 and 370, these  
 o gears are drawn in outline, the simpler type first, the loco-  
 tive type second—both in the locomotive position. The effect  
 the reversing rocker-arm, as  
 forth in Fig. 341, and further  
 icated by the dotted-line eo-  
 tric-settings in Fig. 364, is  
 re shown in the reversed posi-  
 m of the crank, with reference  
 the figure of the eccentrics,  
 Fig. 370. These diagrams are  
 ended to make clear the dis-  
 ction between the two possible  
 rangements of the eccentric-  
 ds, or two ways of connecting  
 e eccentrics to the link—that  
 between “open” and “crossed”

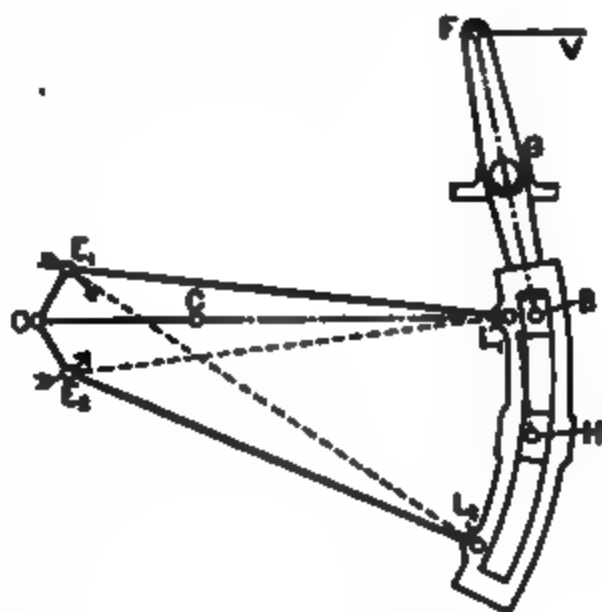


FIG. 370.—The Indirect Link-motion.

ds. During a whole revolution of the shaft, the two rods will  
 part of the time clear of each other, as here, part of the time  
 crossed. The distinction is made by turning the shaft so that

the two eccentric-centers  $E_1$  and  $E_2$ , are toward the link, or the link and a vertical center-line through  $O$ ; and then, for this characteristic position, whether the rods are shown by the full lines, or crossed, according to the dotted lines.

(d) MOVEMENT OF THE VALVE — Anticipating the definition presently to be given, we will now state the manner of the link-motion, as follows:

For any position of the block in the link, intermediate  $L_1$  and  $L_2$ , the motion of the valve is very nearly what is given by a single eccentric with its center on a curve between  $E_1$  and  $E_2$ ; the location of this center  $E$  between  $E_1$  and  $E_2$  is similar—in the geometrical sense of proportionality—to the location of  $B$  between  $L_1$  and  $L_2$ . To make this statement fit exactly, we should designate as  $B_1$  and  $B_2$  the block-rod right in line with the two eccentric-rods, and make the proportionality apply to the location of  $B$  on  $B_1B_2$ .

In Fig. 371 the first thing shown, at I., is the manner

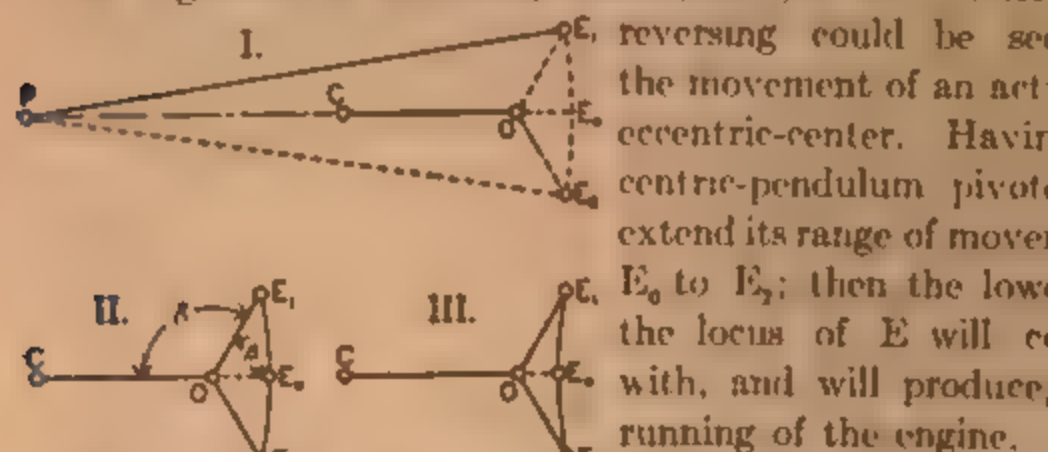


FIG. 371 — The Eccentric-locus.

tion, is shown at II. and III. on Fig. 371 for the respective open and crossed rods. In strict accuracy parabolas, the are well enough approximated by circular arcs through  $E_1$ ,  $E_0$ , and  $E_2$ ; the point  $E_0$  being located, or the length mid-gear radius  $OE_0$  given, by the formula

$$r_0 = r \left( \cos \beta \pm \frac{k}{l} \sin \beta \right); \quad . . . .$$

where  $r$  is the actual eccentric-radius,  $\beta$  is the supplement of  $\delta$  as Fig. 371 II.,  $k$  is half the length  $L_1L_2$  of the link, and  $l$  is the length of the eccentric-rod. The plus sign is for open rods, giving the convex locus in II., or increased lead toward mid-gear; the minus sign and the concave curve are for crossed rods. Note that the second term of the formula,  $(k/l)r \sin \beta$ , is the amount by which the curve departs from the straight line  $E_1E_2$ ; being the fraction of the half-length of this line.

The "adjustment" of the valve-gear, or the position of the link in the block, is defined as follows:

When the block is under the full control of either eccentric—it is, at  $L_1$  or  $L_2$  in Fig. 369, in line with  $L_1$  or  $L_2$  in Fig. 370—the mechanism is said to be at "full-gear," forward or backward. Mid-position is called mid-gear, as indicated above. Any other position is described by giving the fraction got by dividing the half-length of the link into the distance of the block from the middle of the link: thus we speak of half-gear, three-quarters gear, and on.

Positions of the reverse-lever latch on its arc will be almost exactly similar to those of the block in the link: so that the setting of the valve-gear can be equally well defined in terms of this position.

(e) VALVE AND INDICATOR DIAGRAMS.—A sample set of valve-diagrams for a Stephenson gear is given in Fig. 372. The eccentric-

crank is first laid out at  $E_1E_2E_3$  (for the crank at zero), and the locus of  $D$  is made symmetrical to it, with reference to the line  $GH$ . Valve-circles are drawn, for full-gear forward, on  $OD_1$ ; for full-gear backward, on  $OD_2$ ; for mid-gear, on  $OD_3$ ; and for half-gear forward, on  $OD_4$ ;  $OD_4$  and  $OE$  corresponding with a position of block in link half-way between  $L_1$  and the middle, according to the definition just given.

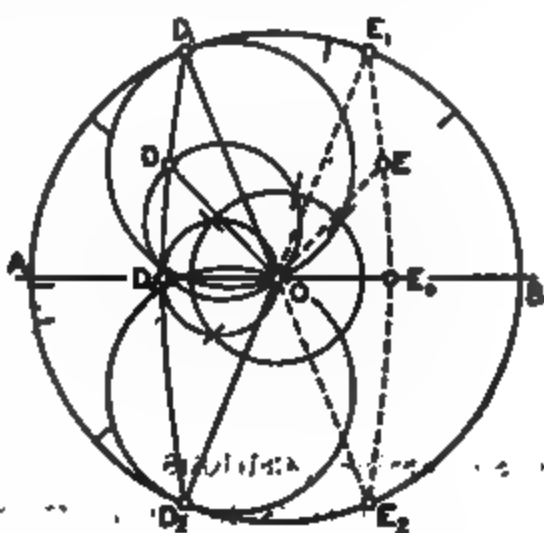


FIG. 372.—Valve-diagrams for the Link-motion.

Several indicator diagrams from a locomotive are reproduced

in Fig. 373. The largest, No. 1, shows the greatest amount of work per revolution that the engine is capable of performing; this work can be done only at very low speeds, and the tractive force, or tractive force at the driving-wheels, due to this steam diagram is so great that it would slip the wheels if friction were not increased

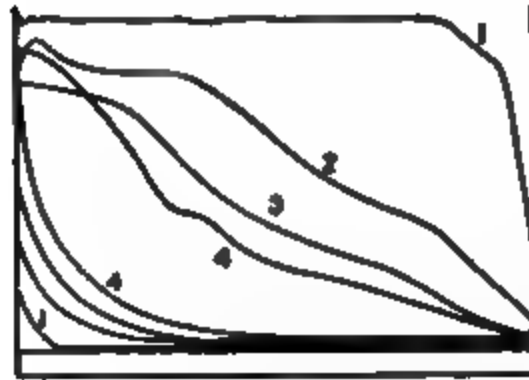


FIG. 373.—Diagrams from a Locomotive.

by the use of sand on the rails. The normal resistance of the train is represented by the straight line  $ab$ , taken at speeds of 35 miles per hour. Slight variations in boiler-pressure and in the opening of the throttle-valve, together with the influence of speed as already illustrated in Fig. 355, account for the differences in the admission pressure, and render impossible

close determination of a cut-off locus like  $QQ$  on Fig. 354.

**PROBLEM 11.** Given  $r$ ,  $s$ ,  $k$ , and  $l$ , and that the rods are open. First set the eccentrics so as to have zero lead in full-gear; then get the eccentric-locus, and draw Zeuner diagrams for full gear both ways, for mid-gear, and for quarter cut-off forward. Show the steam distribution in each case, taking the inside lap to be zero. Good average data are,  $r=2\frac{1}{2}$ ",  $s=1$ ",  $k=8$ ",  $l=50$ ".

### § 51. Kinematics of the Link-motion.

(a) **THE MECHANISM SIMPLIFIED.**—The movement of the valve as produced by the link-motion, is the resultant of a number of component actions. The more important of these are comparatively simple, and can readily be brought to general geometrical expression; but some of the smaller, secondary influences can be investigated only by the method of plotting the whole mechanism in successive positions.

In order to isolate the effect of the eccentrics, we shall use the model shown in Fig. 374, where the movement which they give is transmitted to the link without change. The following description will make clear the working of this mechanism:

the several parts of the framework bear the number 1; the disk turning in a fixed bearing, represents the crank-shaft; the eccentrics are reduced to pins fastened in 2, one projecting to the front, the other behind. These pins work in cross-slots in the slides 3 and 4, after the manner of Fig. 106; and the harmonic motion produced is transmitted by the rods 5 and 6 to the slides 7 and 8. These work in guides on the adjusting-slide 9; and it is to be noted that the joints between 3 and 5, 5 and 7, etc., are not

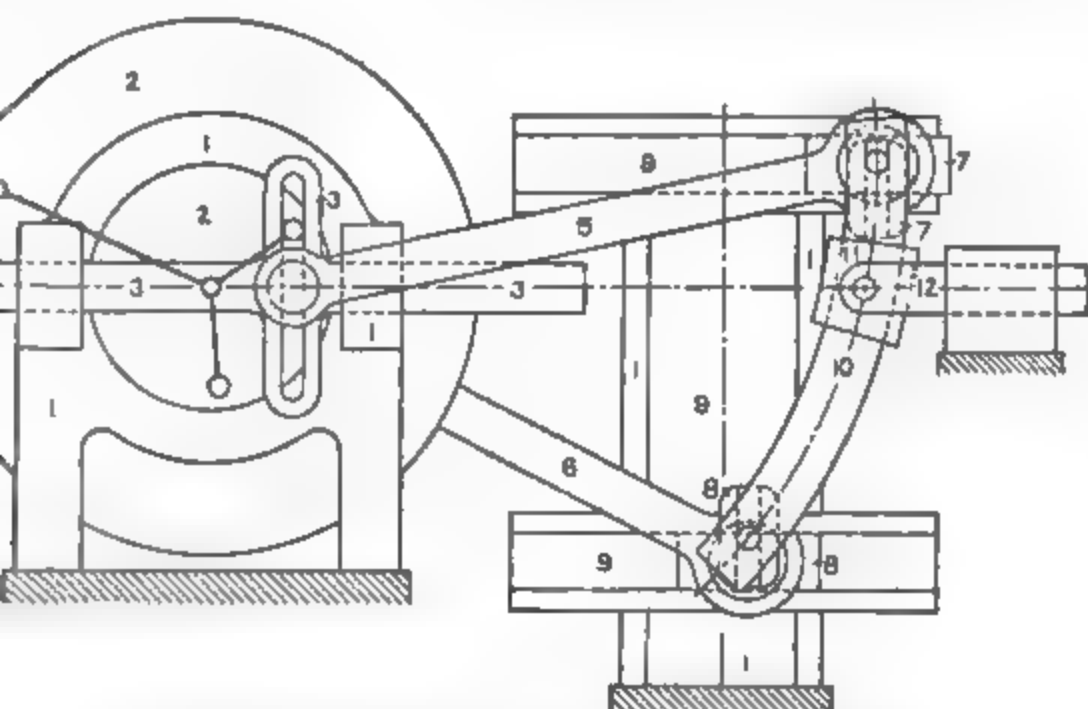


FIG. 374. — Kinematic Model of the Link-motion.

ing joints, but turn only when the gear is "adjusted" by moving up or down. Slides 7 and 8 carry vertical slots in which work rod-pins on the back of the link. The latter is of the simple type, as in Fig. 368; and is supposed to be supported by placing the block 11 upon it in the position proper to any setting

**HARMONIC MOTION TRANSMITTED** — With the model arranged as in Fig. 374, the ends of the link receive harmonic motion exactly as though they travelled on a stroke-line passing through the center of the shaft. In Fig. 375, the ideal mid-position of the link, from which its movement is to be measured, is found by striking the arc with a compass from O, with the rod-length  $l$  as radius. Then the effect of

the arrangement under discussion is to make  $P_1B_1$  equal  $OD_1$  and  $P_2B_2$  equal  $OD_2$ . Calling these travels  $t_1$  and  $t_2$ , the actual travel  $t$  or  $PB$  is a mean between them, in a proportion determined by the position of  $B$  on the link  $L_1L_2$ .

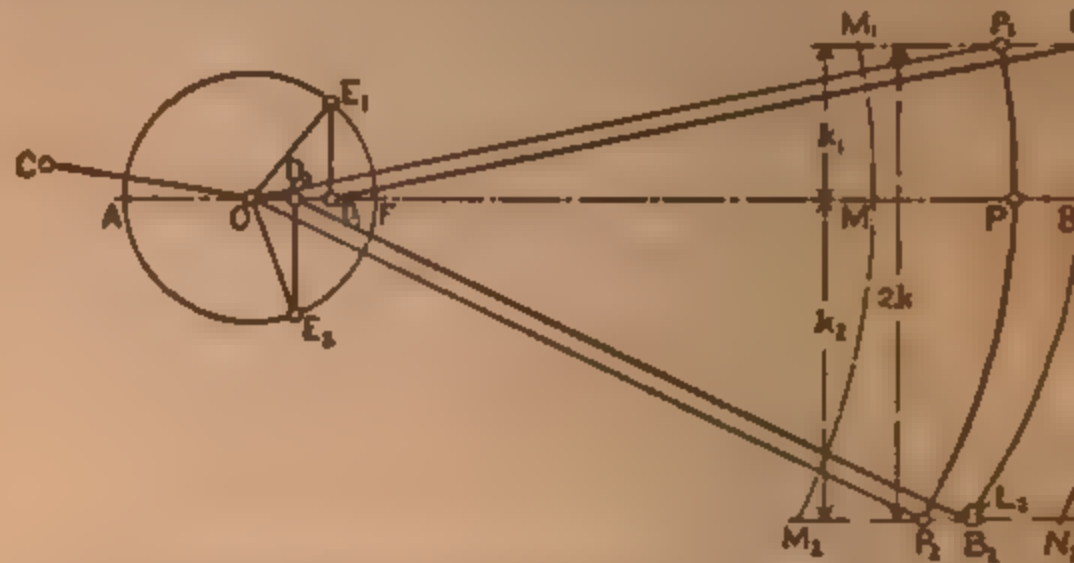


FIG. 375. — Harmonic-motion Component of Valve-travel

the location of  $B$  on  $L_1L_2$ : that is, the difference between  $t_1$  and  $t_2$  is divided in the same ratio as the link is divided by the block  $B$ .

Fig. 376 I. shows an enlarged diagram of the eccentrics; it is evident that by locating  $E$  on  $E_1E_2$  just as  $B$  is located on  $L_1L_2$ , we secure the same value of  $t$  as on Fig. 375. This brings us to the conclusion that the valve-slide, driven by  $B$ , will move as though it were under the control of the single eccentric  $OE$ .

(c) CURVATURE OF THE LINK — In strict accuracy, a true constant proportionality in the interpolation of  $t$  between  $t_1$  and  $t_2$  would be secured only with the straight link outlined in Fig. 375, where a rigid bar pivoted at  $B$  passes through slide-blocks pivoted on the ends of the rods and guided along the stroke-lines  $M_1N_1$  and  $M_2N_2$ ; and it may be remarked that a curved link made on this principle might give a smaller error than does that actually shown on Figs. 374 and 375. But the straight link would have the real fault that the point  $P$ , the center of movement, would change position as the link was moved up and down.

It appears then that only by using for the center-line of the link an arc which has the length  $l$  as radius can the stroke-line of Fig. 375, be kept at a constant position as the valve-gear

ed. The limit-curves  $M_1M_2$ ,  $N_1N_2$ , are, of course, arcs struck at A and F with the radius  $l$ .

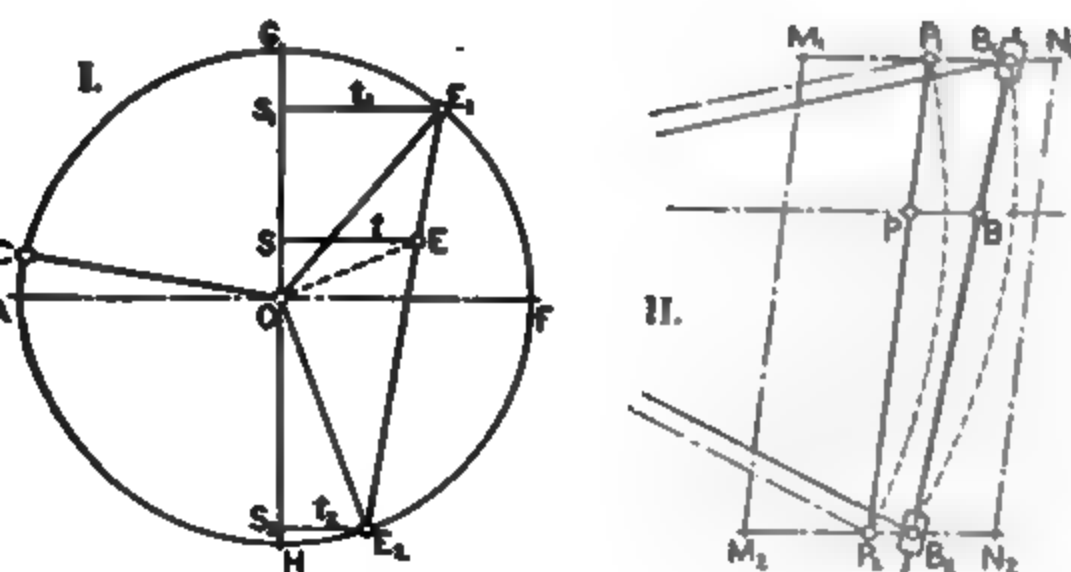


FIG. 376.—The Equivalent Eccentric: The Straight Link.

As regards the effect of this curvature upon valve-movement, note that the angle between the two chords  $BL_1$ ,  $BL_2$  on Fig. will modify slightly the paths of the pins  $L_1$  and  $L_2$  in an actual, and will also have a small influence upon the relation of  $t$  and  $t_2$ ; but with actual proportions, instead of the very short

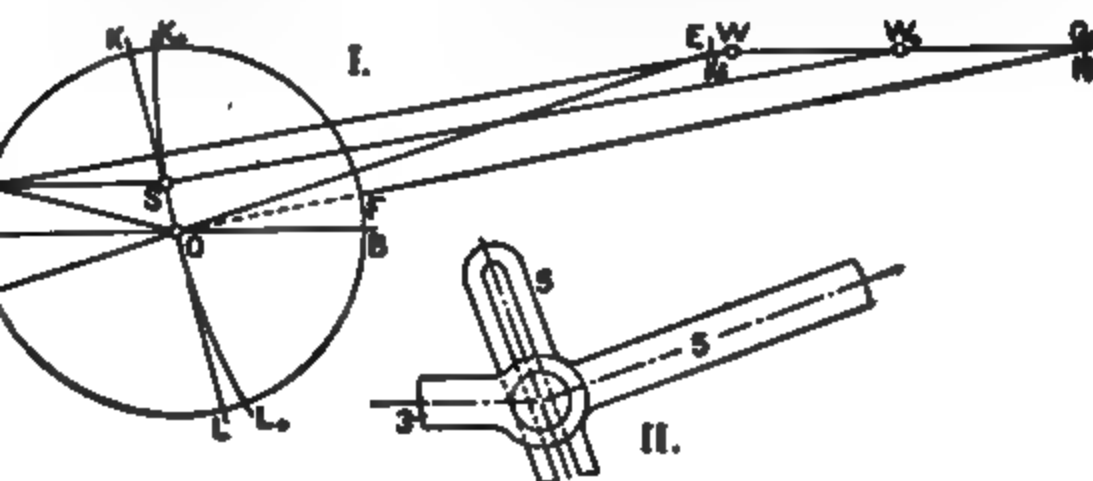


FIG. 377.—Non-central Stroke-line in the Link-motion.

used in these illustrative figures, both these effects will be very small.

(d) INCLINATION OF THE ECCENTRIC-RODS.—The next step is to take into account the fact that the link-pins, as actually driven



by the rods, travel on stroke-lines which do not pass through the shaft-center. This condition is shown by the diagrams in Fig. 377, which is similar to Fig. 362. Here  $M$  and  $N$  are the same points as  $M_1$  and  $N_1$  on Fig. 375, the dead-centers being at  $E$  and  $G$ . Continuing to neglect the effect of the swing of the eccentric-rod, we get that of its mean position. Taking a part of the model, Fig. 374, and putting it in the form shown in Fig. 378, where the slot for the eccentric-pin is carried by the crank, we find that the rod is always to be at right-angles to the center-line of the crank, the motion of the rod being guided along a horizontal line by the

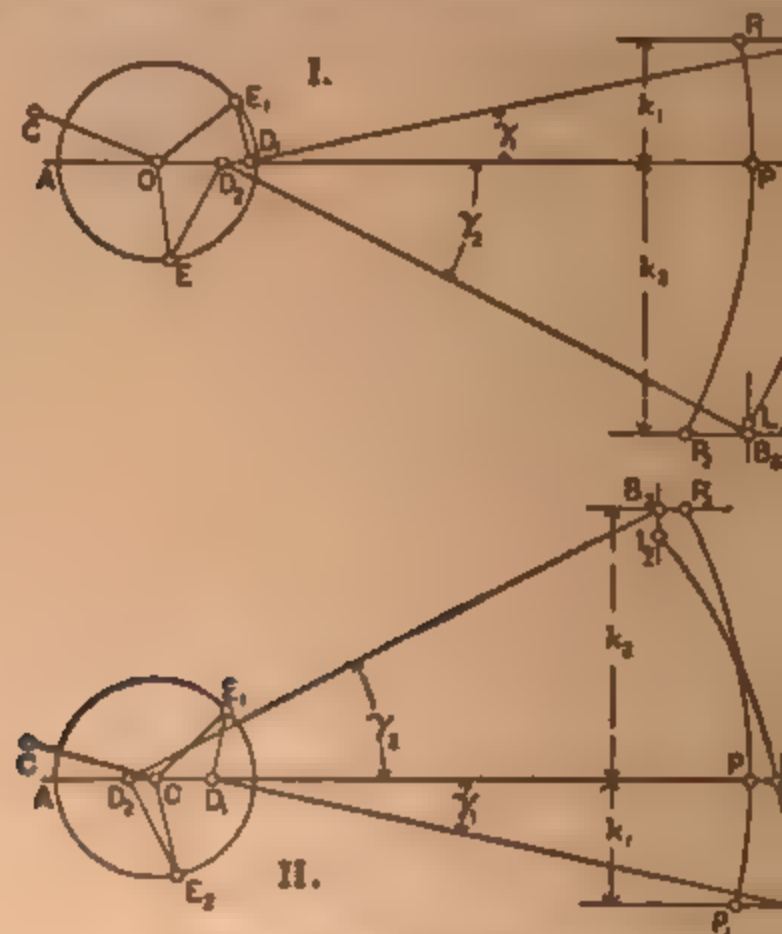


FIG. 378.—Effect of Rod-inclination.

Diagrams for this motion, similar to Fig. 375. Fig. 378, I. for open rods, II. for crossed rods. The rods are now perpendicular to the rod-lines  $DB$  instead of being perpendicular to the line  $AP$ ; and with the exaggerated proportions here shown, the valve-travel is very materially modified by this important difference between the two rod-arrangements.



is apparent, in that the travel is increased with open rods, decreased with crossed rods, from that in Fig. 375: and this result is general, holding true all the way round the circle.

Fig. 379 gives enlarged eccentric diagrams for the two cases, analogous to Fig. 376 I. The reference-lines  $K_1L_1$ ,  $K_2L_2$ , are drawn at right-angles to the mean rod-line, like  $KL$  on Fig. 377. In each case, comparing with Fig. 378,  $E_1S_1 = OD_1 = P_1B_1$ ,  $E_2S_2 = P_2B_2$ ; and  $E$  divides  $E_1E_2$  in the same ratio that  $B$  does. Then joining  $S_1S_2$  and drawing  $ES$ , we have in  $ES$  a proportional between  $E_1S_1$  and  $E_2S_2$ , just as  $PB$  is a proportional

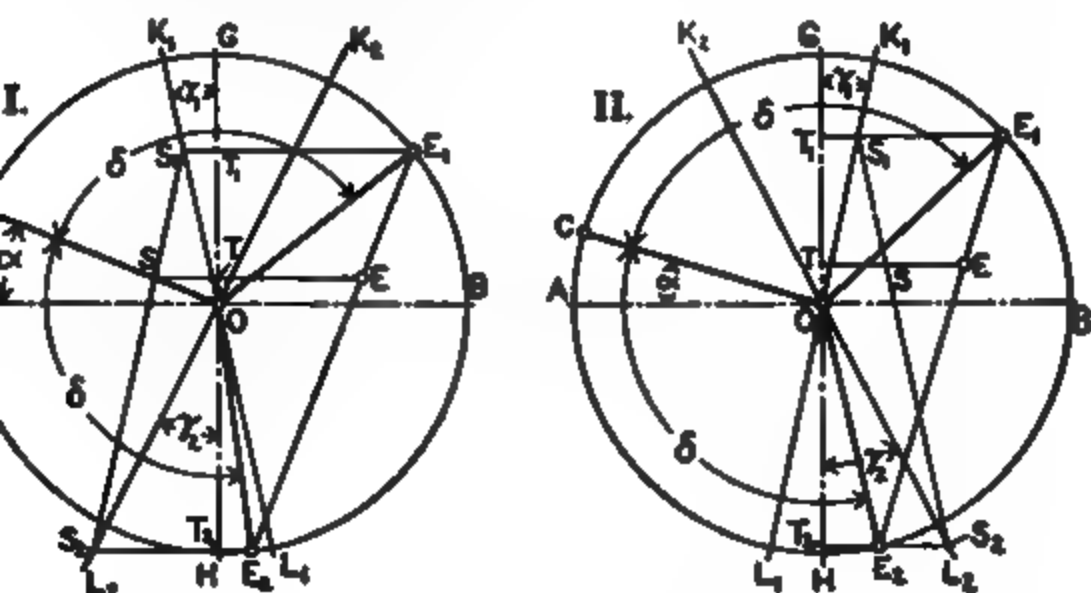


FIG. 379.—Eccentric Diagrams.

between  $P_1B_1$  and  $P_2B_2$ . It will be noted that each  $ES$ -line is made up of two parts; the portion  $ET$  corresponds to harmonic motion, while  $TS$  shows the effect of the slant of the rods.

(e) DETERMINATION OF EFFECT OF ROD-SLANT.—As the first step in the development of a general graphical expression for the travel-component  $TS$ , we must find a simple way of representing the movement of the slide in the case of Figs. 362 and 363, carrying out to a definite conclusion the approximation suggested at  $OE_0$  in Fig. 362. This is done in Fig. 380, where  $OE$  is the actual eccentric,  $OW_0$  the line of mean rod-slant, and  $KL$  the reference-line in the common circular diagram. First, put the eccentric on its dead-center line  $OW_0$  at  $OE_1$  (referring to Figs. 362 or 377, we see that this will be the line of both dead-

centers if swing of the rod is disregarded): then the travel  $S_1E_1$  is the same as would be produced by a radius  $OF_1$ , got by drawing

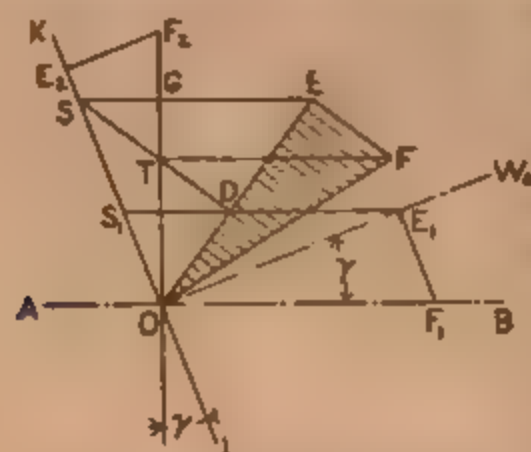


FIG. 380 — Equivalent Eccentric with Non-central Stroke-line.

$E_1F_1$  perpendicular to  $OE_1$ , and driving a slide along the line  $OB$ . This suggests the relationship exemplified at  $OEF$ , where the angle  $EOF$  is, of course, equal to  $\gamma$ , and we must now prove that, in general,  $TF$  is equal to  $SE$ .

Geometrically, draw  $SD$  and  $EF$  perpendicular to  $OE$  (that  $D$  falls on  $S_1E_1$  is a mere coincidence), and  $FT$  parallel to  $SE$  from the intersection  $T$ , to cut off the length  $EF$ ,

then to show that  $EF$  is constant and  $DOF$  equal to  $\gamma$ , note that the triangles  $STG$ ,  $OEG$  are similar, having respective sides at right angles; then

$$ST : OE :: SG : OG;$$

and  $EF = ST$  by construction; wherefore

$$\frac{EF}{OE} = \frac{SG}{OG} = \tan \gamma.$$

In this demonstration, we really derive an equivalent eccentric which makes  $TF = SE$ . Using trigonometrical relations, we could prove the equality directly for the relation assumed above; but it is aside from present purposes to multiply proofs.

(f) COMBINED EFFECT OF THE TWO RODS. — In Fig. 381, we make  $E_1F_1$  equal  $r \tan \gamma_1$ ,  $E_2F_2$  equal  $r \tan \gamma_2$ , taking the angles as on Fig. 379: then these extra eccentric-components are projected upon the horizontal travel-lines  $E_1T_1$ ,  $E_2T_2$ , giving the extra-travel  $E_1D_1$ ,  $E_2D_2$ , at the respective ends of the link, these distances being necessarily the same as  $T_1S_1$ ,  $T_2S_2$  on Fig. 379. Next we join  $D_1D_2$ , and cut off the length  $DS = t$  from the horizontal through  $E$ ; and we see that the eccentric  $OF$ , of which  $SD$  is the horizontal projection, is the resultant of the original eccentric  $OE$  and of the extra component  $EF$ ; which last, as must now be parallel

the crank  $CO$  and a proportional between  $E_1F_1$  and  $E_2F_2$ , determined by the ratio of  $E_1E$  to  $EE_2$ .

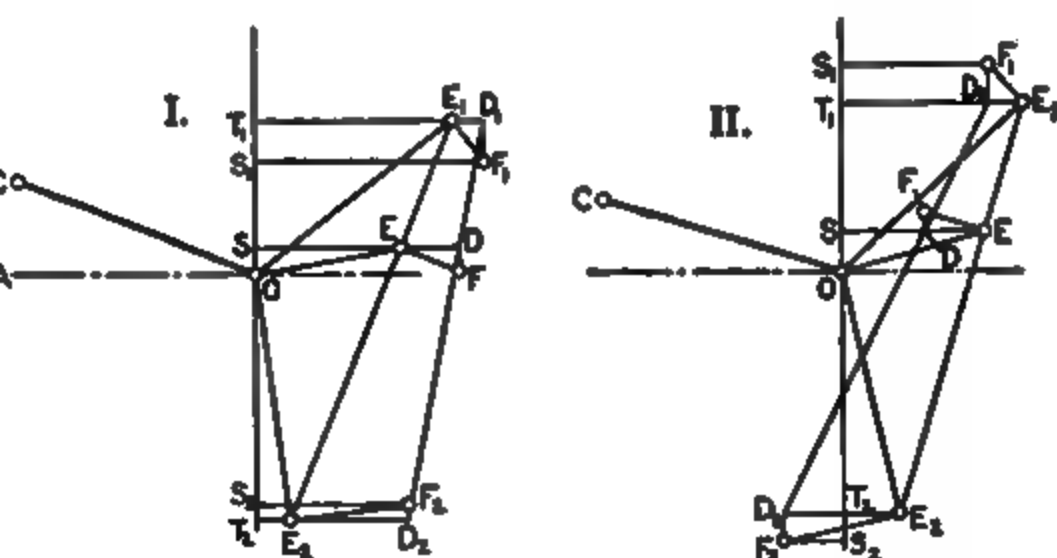


FIG. 381.—Rod-effect by Equivalent Eccentrics.

Just as the resultant of  $OE_1$  and  $OE_2$  is  $OE$ , drawn from the center  $O$  to a proportional point  $E$  on the line  $E_1E_2$ , so also can the resultant-effect  $EF$  of  $E_1F_1$  and  $E_2F_2$  be found by a graphical combination. This is worked out in Fig. 382, where the rod-

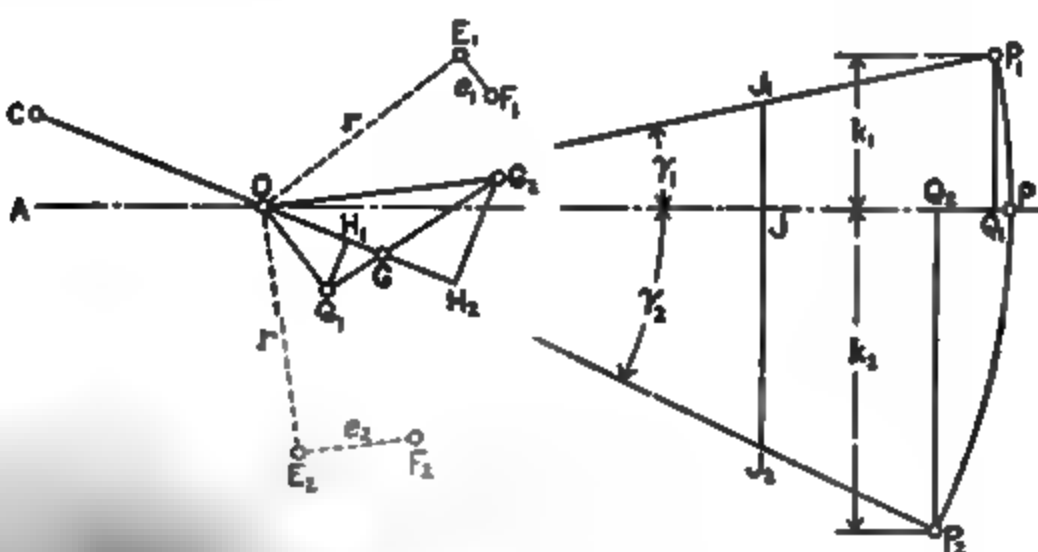


FIG. 382.—Combination of the Rod-effects.

and  $\cdot$  are transferred to the center  $O$  (being as to make a figure of better size). be most conveniently got by drawing on in its ideal mid-position, at  $OP_1P_2$ .

then measuring off  $OJ=r$  and drawing the vertical  $J_1J_2$ : evidently,  $JJ_1=r \tan \gamma_1$ ,  $JJ_2=r \tan \gamma_2$ . Here  $OJ$  is  $r$  to the enlarged scale, so that  $OG_1$  and  $OG_2$  are given directly by  $JJ_1$  and  $JJ_2$ . These two lines are, of course, perpendicular to their respective eccentric radii,  $OE_1$  and  $OE_2$ ; therefore each makes with the crank-line an angle equal to the angle of advance of the eccentric to which it belongs.

Now join  $G_1G_2$  and let the crank-line intersect it at  $G$ : then since  $OG$  bisects the angle  $G_1OG_2$ , it divides the base  $G_1G_2$  of the triangle into segments proportional to the sides. This means that  $G$  is located on  $G_1G_2$  just as  $J$  is on  $J_1J_2$ , and very nearly as  $P$  on  $P_1P_2$  or  $E$  on  $E_1E_2$ ; wherefore the projection of  $OG$  upon the horizontal stroke-line will be a mean between those of  $OG_1$  and  $OG_2$ , according to the ratio of  $PP_1$  to  $PP_2$ , just as  $ED$  is an intermediate between  $E_1D_1$  and  $E_2D_2$  on Fig. 351.

(g) EXPRESSION FOR THE COMBINED ROD-EFFECT.—One approximation is included in the statement just made, that of assuming

$$PP_1 : PP_2 = JJ_1 : JJ_2.$$

To get a formula for the length  $OG$ , we avoid excessive complexity only by the further approximation of taking  $k_1/l$  for  $\tan \gamma_1$ ,  $k_2/l$  for  $\tan \gamma_2$ , or of letting

$$e_1 = \frac{k_1}{l}r, \quad e_2 = \frac{k_2}{l}r.$$

Then since  $G_1OG = GOG_2 = (90 - \beta)$  (see Fig. 371), we have

$$OH_1 = k_1 \frac{r}{l} \sin \beta, \quad OH_2 = k_2 \frac{r}{l} \sin \beta,$$

$$H_1H_2 = (k_2 - k_1) \frac{r}{l} \sin \beta.$$

To simplify, let  $n$  be the "fraction of gear" as in § 50 (d), or the ratio which the distance of  $P$  from the middle of the link bears to the half-length  $k$ , Fig. 375: then  $k_1 = (1-n)k$ ,  $k_2 = (1+n)k$ .

Now  $H_1G$  is the fraction  $\frac{k_1}{2k}$  or  $\frac{1-n}{2}$  of  $H_1H_2$ : and for  $OG$  or we have

$$e = OH_1 + H_1G = \left(1 - n + \frac{1-n}{2} \times 2n\right) \frac{k}{l} r \sin \beta$$

$$= (1 - n^2) \frac{k}{l} r \sin \beta. \quad (314)$$

mid-gear, or  $n=0$ , this is identical with the second term of (313); and from the form of the variable factor  $(1-n^2)$ , we see that the E-curve, where  $e$  is laid out on a base proportional to  $\sin \beta$ , is a parabola.

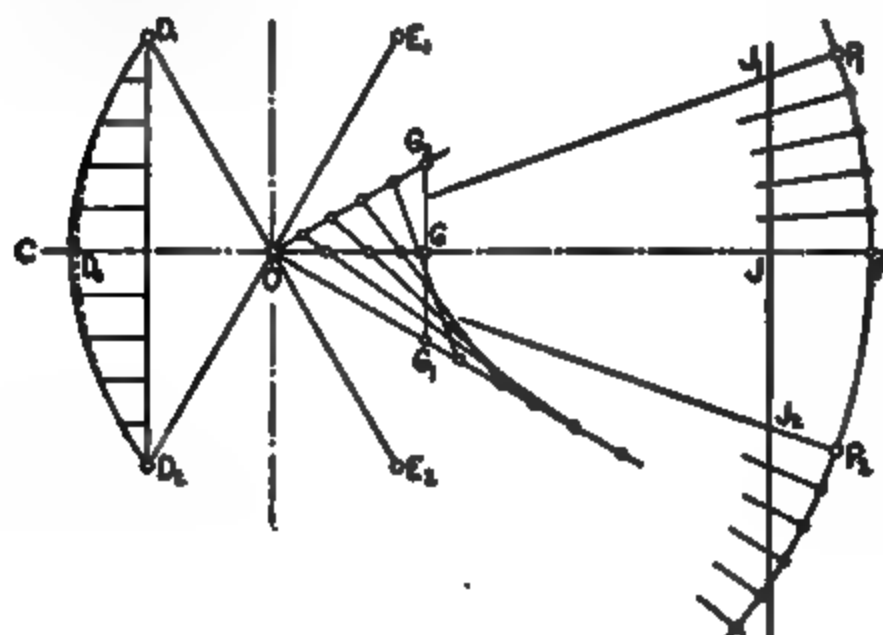


FIG. 383.—Construction of the Locus of E.

In Fig. 383 the construction for  $G$ , from Fig. 382, is made for a number of positions, and the results plotted. The crank is put lead-center, which places the  $OG$ -lines in the most favorable position. Only half the curve, from mid-point to one extreme, need be determined, as the halves are symmetrical: the figure drawn is not quite consistent, since constructed values of  $OG_1$  and  $OG_2$  are interchanged in laying them out; but this does not affect the results.  $OG$  is found as of double size, then reduced and laid off as  $D_1D_2$ , this line being used instead of  $E_1E_2$ , merely to avoid confusion. Finally,  $OE_0$  or  $OD_0$  is got by Eq. (313) and the locus drawn in dotted line. The difference between the two curves is a measure of the effect of assuming  $k/l=$

$\tan \gamma$ ; and it appears that even here this causes a very small error which would disappear with practical proportions.

This completes our study of the principal movement of the link-driven slide. Taking up next the secondary actions, we consider first the effect of the angular swing of the rods about their lines of mean slant, then the irregular movements at the link especially the slip of the block in the link.

(b) OSCILLATION OF THE RODS.—The effect of this element of disturbance can be closely enough determined by the method of Fig. 357, applied through Fig. 377, disregarding the fact that the rod-pins in the link do not really travel in straight-line paths parallel to the stroke-line of the valve. An example is worked out in Fig. 384, where the proportions used are

$$\frac{l}{r} = 10, \text{ as against a usual value of 20 or more;}$$

$$\frac{k}{r} = 2.8, \text{ the range of practice being from 2.5 to 3.5.}$$

Assuming that the leads are made equal in full gear, we pass the arc KL, with  $l$  as radius, through points found by projecting the zero positions of  $E_1$  and  $E_2$  over to the vertical center-line. With the one eccentric in full control, this curve gives a complete solution of the problem, for full gear. The more general case is represented by half-gear forward, for which are derived and plotted the results shown at II.

The curves  $K_1L_1$ ,  $K_2L_2$ , are drawn from centers on the lines  $OP_1$ ,  $OP_2$ . Angle-scales for  $E_1$  and  $E_2$  are laid out, the first numbered outside the circle, the second inside; by connecting points with the same number, we should get successive positions of the eccentric-span  $E_1E_2$ . For each position, the horizontal distance between the particular KL-curve and its straight line is measured for both eccentrics; and a mean between the two in proper proportion—here one-fourth of the way from the first to the second—is laid off from the end of the Zeuner ordinate at II, with due regard to direction, each resulting point being marked by a small circle. The latter figure is made double-size, to show up better.

THE CHARACTERISTIC EFFECT of this action is well brought out by this example. Just as in full gear, though not quite to the degree, eccentric-displacement toward the right gives the port-opening, and should therefore control admission to the end. Keeping in mind the change in the characteristic of the mechanism, this conforms to the statements in (a). Further, valve-movement is no longer symmetrical with respect to the diameter OD, but there is later cut-off when the ec-

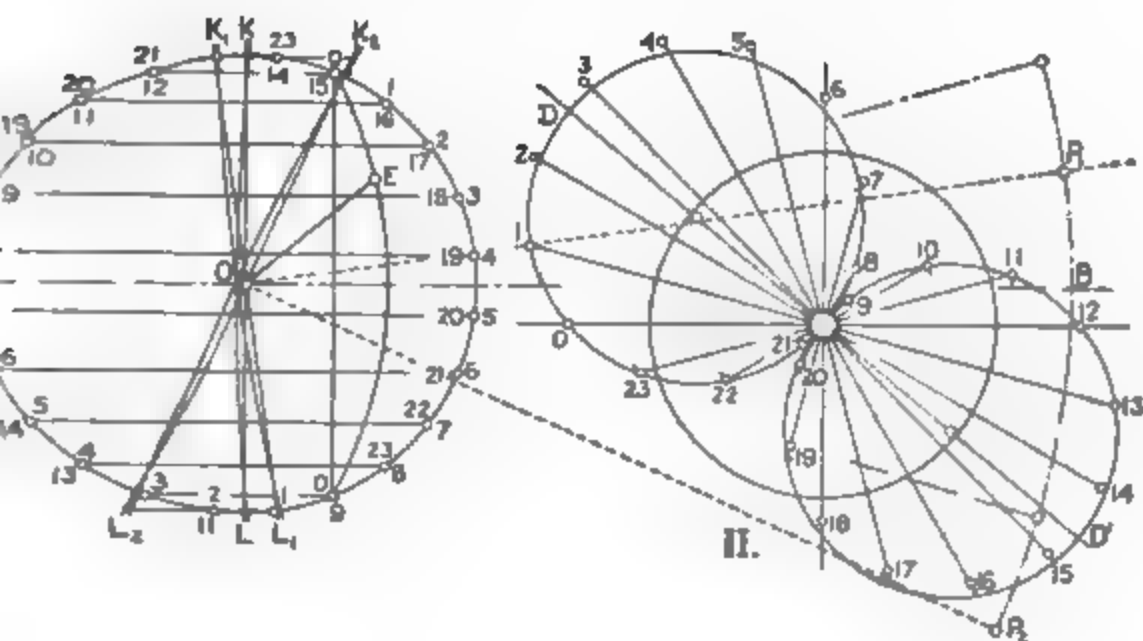


FIG. 384.—Diagrams of Rod-effect.

tric is to the right than to the left. Then with a direct valve directly driven, or with an indirect valve and reversing-rocker, the rod-effect will tend to equalize cut-offs. With the usual arrangement in American practice, however, this effect increases the non-symmetry of the steam-distribution: but when the rods are made long enough to keep the locus of E reasonably near the straight line  $E_1E_2$ , the effect of rod-swing becomes very small.

The same method could be used for any other E-point: and inspection of the figure will show that no great error would result from using the full-gear curve KL instead of the true reference-curve in any case. Then we have only to study the effect of rotating the line  $E_1E_2$  about O, giving proper relative weights to the disturbances. The drawing horizontals from its two ends across the center-line.

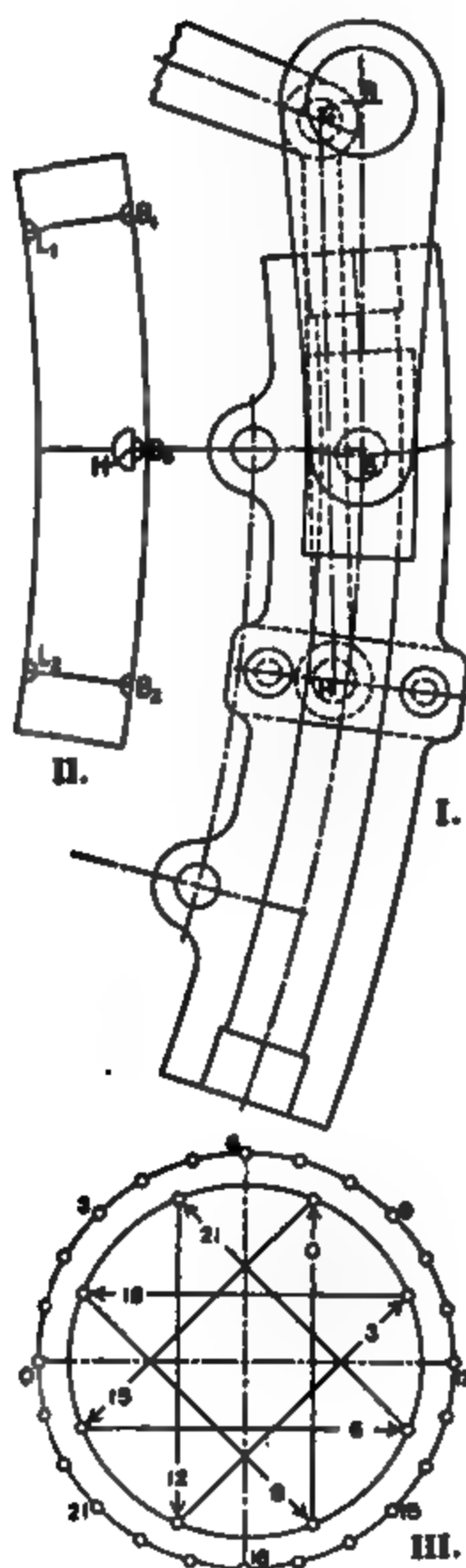


FIG. 385.—The Link and Eccentric Diagram.

At mid-gear, the disturbances due to the rods will be again symmetrical with respect to the OD-line, as at full-gear: but right-hand travels will be lengthened, those toward the left shortened.

(j) EFFECTS OF SLIP AND OFFSET.—It is not practicable to fasten the block in the link, in the position belonging to each setting of the gear: in the locomotive arrangement, the block-pin moves on a circular arc, at the end of the rocker-arm, or determined by a hanger-link as in Fig. 366; when the gear is direct-driving, this pin is either guided in a straight line by a slide, or in an arc of long radius by guiding-rods. Whatever the arrangement, its path is generally not that of a point on the link; and the resulting slip of block in link will have the effect of varying, within the revolution, the center of the equivalent single eccentric that drives the valve. Further, with the form of link shown in Fig. 367, and used almost without exception in American locomotives, the fact that the rod-pins are offset from the center-line introduces another complication into the movement, as the link swings away from its normal inclination. This whole action can best illustrated and explained

and  amply.



The link, Fig. 385 I., belongs to a valve-gear having the same proportions as that in Fig. 384, except that the rods are twice



FIG. 386.—Link-positions.

long, making  $l/r=20$ , and the link-curvature is correspondingly s. in all its proportions, this example represents average practice ry fairly. The link is drawn in its ideal mid-position for full-

gear forward, all the arcs being struck from the center of the shaft (the main axle).

Fig. 385 II. shows a template, to be made of stiff paper or thin sheet-metal, carrying all essential proportions and dimensions of the link, and used in drawing the diagrams in Fig. 386.  $L_1$  and  $L_2$  are the link-pin centers,  $L_1L_2$  the link-arc through them, with the rod-length  $l$  as radius;  $B_1B_2$  is the center-line of the slot, struck from the same center as  $L_1L_2$ , hence with a longer radius. The points  $B_1$ ,  $B_2$ , are determined by radial lines through  $L_1$  and  $L_2$ ; they are the ideal driving-points for the respective full-gear settings. The hanger-pin is usually set a little back of the center of the slot, and is here on the straight line through  $B_1$  and  $B_2$ .

In Fig. 385 III. are given the eccentric-positions for which the diagrams in Fig. 386 are drawn: vertical, horizontal, and mid-quadrant positions of the line  $E_1E_2$  are taken as representative, and the numbering corresponds with that in Fig. 384. The method employed in laying out Fig. 386 is as follows:

For any particular crank-angle, as 6, arcs are struck from the respective eccentric-centers, giving loci of  $L_1$  and  $L_2$ ; from the suspension-pin  $G$ , with  $GH$  as radius, is drawn another arc on which  $H$  must lie; and by trial the position of the template is found that will place the three points  $L_1$ ,  $L_2$ , and  $H$ , each on its proper curve. Then an arc from the rocker-center locates the block-pin, at  $R$ , and the distance  $RB$  shows the slip of the block, from its normal position  $B$ . In each figure, the vertical through  $P$  is the reference-line for the movement of  $L$ ; while  $Q$ , at the distance  $LB$  from  $P$ , fixes the similar center-line for  $B$ .

We note at once that the slip is least when the line  $E_1E_2$  is vertical, at 0 and 12, greatest when this line is horizontal, at 6 and 18, or when the link makes the greatest angle with its mid-position, and that inclination of the link, by inclining the line  $LB$  from the horizontal, makes the distance  $BQ$  differ from  $L-P$ ; (by  $BQ$  and  $L-P$ , and by  $R-Q$ , we mean the horizontal distances from  $B$ ,  $L$ , or  $R$  to the reference-line designated by  $PQ$ ). Around 6, the slip more than compensates for the shortness of  $B-L$ , so that the travel-distance  $R-Q$  greater than  $L-P$ ; at 18 the shorter slip combines with the slant-effect, so as to make  $R-Q$  greater than  $L-P$ .

(k) **LOCATION OF THE HANGER-PIN.**—From examination of Fig. 386 we see that, in the matter of slip and its effect, a great deal depends upon the location of H on the link. Since the travel is nearly the same at 0 and 9 and at 12 and 21, cut-off will occur at about 9 and 21 in full gear. In this example—direct valve with rocker—crank-end admission takes place from 0 to 9, head-end from 12 to 21. To equalize cut-offs, therefore, a certain value of R-Q should be reached sooner near 9 than near 21: and the fact that R-Q, decreasing, has reached a smaller value at 9 than at 21 shows a tendency thus to equalize. Putting H forward, upon the slot-curve  $B_1B_2$ , would increase the inequality; moving it back toward  $L_1L_2$  would at first equalize the travels, then reverse the inequality, because the slip has so much the greater effect at 9.

A complete investigation of the effect of slip and offset, for different gear-settings, would be quite an extensive operation. Having illustrated the method to be used, we shall not go any farther into this very special problem: but it may be well to remark that the location actually used for H is about as here given.

The movement of the link is shown by grouping together at II., in Fig. 386, all the positions of the arc  $L_1L_2$ ; and it appears that the position of the link-template could be found, accurately enough for all practical purposes, by taking the pin-travels for  $L_1$  and  $L_2$  from a diagram like Fig. 384 I. with straight reference-lines, and laying them off horizontally from the mid-position locus  $PP'$ . Short straight lines, parallel to the nearest part of this arc, would replace the true loci of  $L_1$  and  $L_2$  as drawn in I., and would save the trouble of repeatedly using the long radius  $l$ .

In the marine-engine gear, Fig. 368, the hanger-pin is in line with the forward-running rod-pin, and there will be little slip when the forward eccentric has predominant control: the idea being, to favor the more usual condition of operation, at the expense of that which exists only occasionally.

(l) **AS THE RESULT OF THIS DISCUSSION** of the secondary disturbing actions, under Figs. 384 to 386, we may conclude that, with usual proportions, departures from the harmonic motion due to the equivalent single eccentric will be relatively small. For most purposes, the simple valve-diagram, as in Fig. 372, gives

adequately estimate results. But it is well to have an idea of the character of the resulting motion, and to be able to determine their effects with the minimum of work. Of course, the ultimate determination is that made by actually plotting the movement as in Fig. 376.

IN NON-STEPPED ECCENTRICS IN THE STEPHENSON GEAR —In order to study the arrangement completely, we must consider the most general type of arrangement. The simplest case, and the frequently applied in practice, is represented by Fig. 28, where the eccentric-radii are of the same length but the angles  $\beta_1, \beta_2$  are unequal. Reviewing the derivation of the eccentric locus and considering Figs. 376, 379, and especially 381 and 382, it is not hard to see that the point  $E_4$ , the equivalent center where the link is at right angles, is really determined by the bisector of the angle  $E_1E_2$ . In the usual arrangement, this bisector coincides with the crank-line; in Fig. 387 it makes a small angle with the crank; but after it has been determined, the E-curve is to be used, and any particular center located, just as in Fig. 371. The essential point of this argument is, that the triangle  $E_1OE_2$  is the important determinant at the shaft, the position of the crank relative to this triangle having no effect upon the absolute movement of the valve, but affecting only its movement with reference to the crank and the piston.

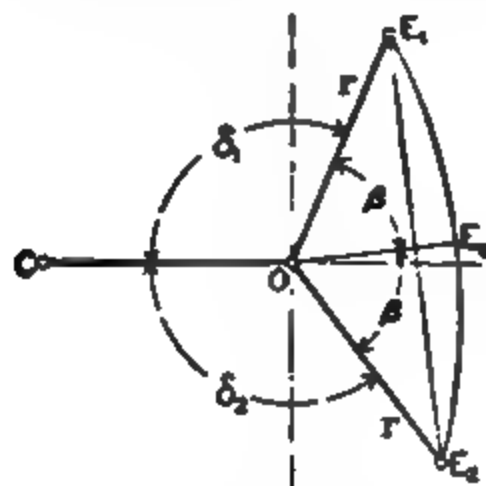


FIG. 387.—Equal Eccentrics at Unequal Angles.

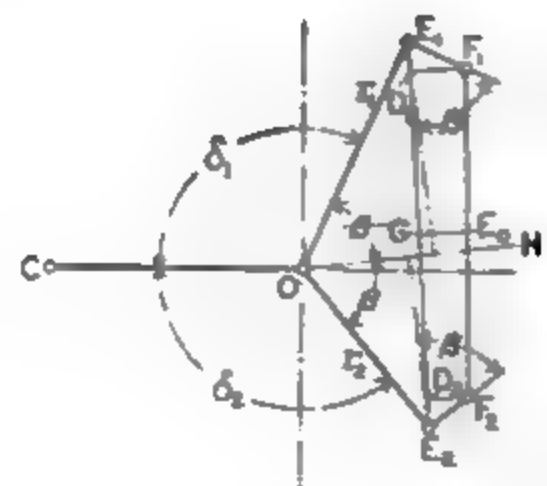


FIG. 388.—Both Radii and Angles Unequal.

The most general case, in Fig. 388, has the eccentric-radii unequal as well as the angles. Under the conditions of Fig. 376,

the mid-gear point  $G$  on the straight locus would bisect the length  $E_1E_2$ ; and the corresponding rod-slant effect  $GE_0$  is a mean between  $E_1F_1$  and  $E_2F_2$ , or between  $D_1F_1$  and  $D_2F_2$ . Using  $k/l$  for  $\tan \gamma$ , as under Fig. 382, we have

$$E_1F_1 = \frac{k}{l}r_1, \quad E_2F_2 = \frac{k}{l}r_2.$$

Further, disregarding the error due to the fact that  $OG$  no longer quite bisects  $E_1OE_2$ , we get  $GE_0$  nearly enough by taking

$$GE_0 = \frac{D_1F_1 + D_2F_2}{2} = \frac{k}{l} \frac{r_1 + r_2}{2} \sin \beta. \quad \dots (315)$$

Practically, then, we have only to join  $E_1$  and  $E_2$  by a straight line, measure off perpendicularly from the middle of it the distance given by (315) to get  $E_0$ , and draw an arc through these three points.

Another departure from symmetry of arrangement is seen in Fig. 454, where the stroke-line of the block-pin—as most nearly represented by a horizontal straight line—does not pass through the centre of the axle. This condition of affairs is quite often found in locomotives, both in the valve-gear and in the main engine-mechanism; but the amount of offset is small compared with the rod-lengths. It can be taken into account by the methods of Fig. 380, which will slightly change the points  $E_1$  and  $E_2$  of the  $E$ -locus from the actual centers. Practically equivalent is the assumption that the stroke-line passes through the shaft-center but is no longer horizontal; then the method of Fig. 342 can be used in drawing the valve diagrams.

(n) OTHER LINK-MOTIONS.—Two variations upon the Stephenson gear are outlined in Fig. 389, both drawn in the characteristic position for the stationary engine, in order to bring out the fundamental relations in that case—although the gears have been used mostly on locomotives. In the Gooch mechanism, at I., the link is kept in a constant suspension, and the driving-rod  $VB$  shifts: this makes necessary a reversal of the curvature of the link. In the Allen gear, both link and rod are moved, by a double suspension—

lever: the link is straight so that the center of travel of the block varies, but this is compensated by the effect of the slant of the rod VB with the result that V has a practically constant mid-position. The proper proportioning of

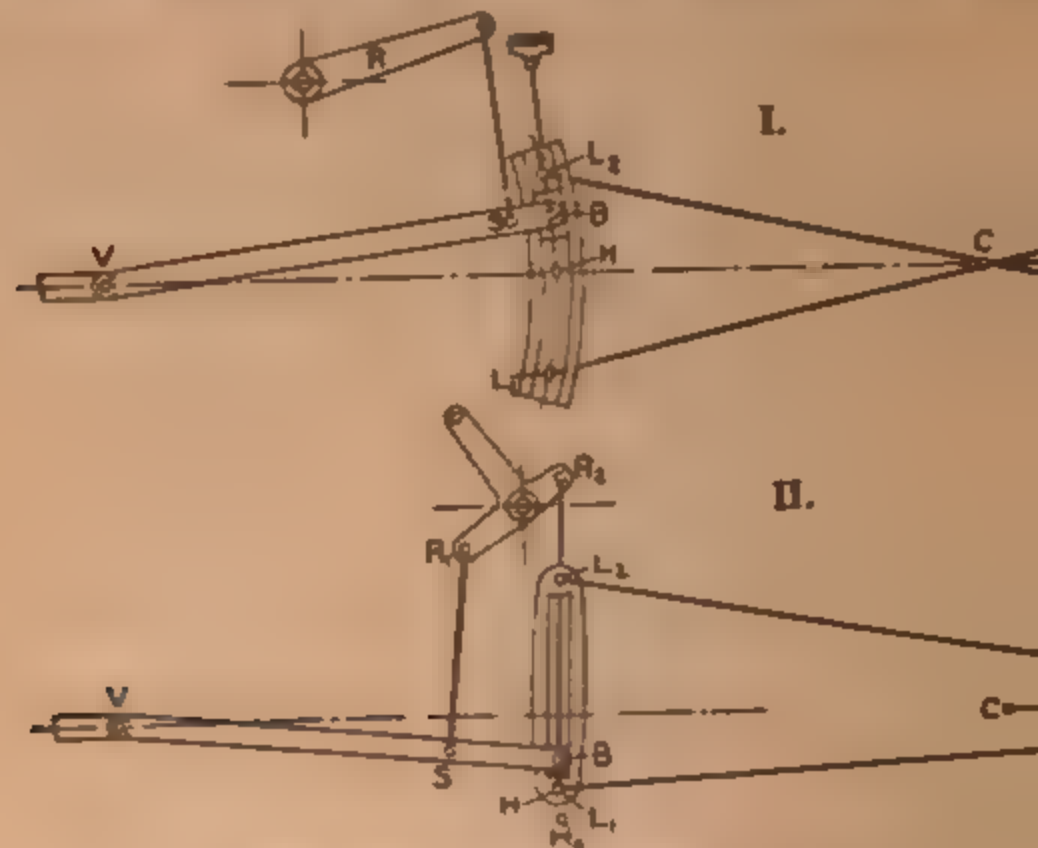


FIG. 389 — Goosch and Allen Link-motions.

suspension-arms is an important point in the design of the gear.

The rod VB, in either case, remains at a practically constant inclination as the shaft rotates; and transmits horizontal motion without change, as do the eccentric-rods in Fig. 374.

It will be noted that, for the engine-position here shown with open rods, the forward eccentric (the one for right rotation) is connected to the lower end of the link; so that the rods are crossed when the crank is on the head-end dead-center. The general rule stated in § 50 (c) applies equally well to this case, however, as appears from Fig. 389 II.

(d) ANALYSIS OF THE GOOSCH GEAR — Here the link has the movement which an equivalent Stephenson link would

mid-gear. Laying out the two actual eccentrics at  $OE_1$  and  $E_2$  in Fig. 390 I., we find the equivalent eccentrics driving the links of the links after the manner of Fig. 380, by measuring off

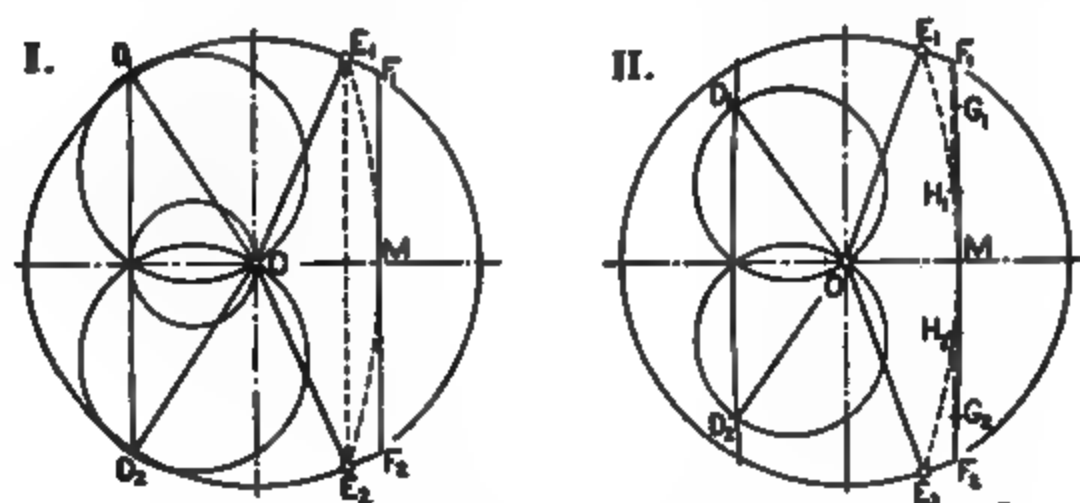


FIG. 390.—Eccentric Diagrams for Gooch and Allen Gears.

equal to  $r \tan \gamma$  or to  $k/l \times r$ : or we may draw the Stephenson curve  $E_1ME_2$  and the line  $F_1F_2$  tangent to it, determining the length of  $E_1F_1$  and  $E_2F_2$ . Then the horizontal movement of any point on the link will be the same as that of a similarly located point on the line  $F_1F_2$ . The full-gear valve-circles are drawn  $OD_1$  and  $OD_2$ , symmetrical with  $OF_1$  and  $OF_2$ .

That the lead will be constant with this mechanism can also be seen, very simply, by the fact that  $V$  will not move as the block moved up and down in the link, when the latter is at the dead-center position, as in Fig. 389 I.

(p) ANALYSIS OF THE ALLEN GEAR.—In determining the locus of the equivalent single eccentric for this case, we use the methods of Figs. 381 to 383: worked out exactly, it would be more complex than for the Stephenson gear, because the driving-point for the block is off the stroke-line  $OV$ . But as a sufficient approximation, we find the extra eccentric-effect due to the mean slant of the controlling eccentric-rod, when the link is at the full-gear position as in Fig. 389 II., and lay this off at  $E_1F_1$  and  $E_2F_2$ . With the mid-gear radius  $OM$ , found by the usual formula, we have three points through which to pass an arc: and the usual rule







of proportionality to block-position is followed in locating the E-point on  $F_1F_2$ .

In the example shown, we have what may be called a "partial-gear" link, because the block cannot move all the way out to the full-gear position. The effect is to limit the eccentric locus at  $G_1$  and  $G_2$ ; and it is for these realized extreme positions that the valve-circles are drawn in Fig. 390 II.

### § 52. The Radial Valve-gears.

(a) UNDER THIS TITLE are comprised a number of mechanisms which give practically harmonic motion to the valve and vary the steam-distribution in the same general manner as the shifting-eccentric or the link-motion. They usually employ only one eccentric, in a valve-gear that can reverse the engine; and in one case, in the Joy gear, there is no eccentric at all in the ordinary sense. The two gears most extensively used, the Walschaert and the Joy, will now be described and discussed; and will serve as typical examples, illustrating the working principles of the whole class.

(b) THE WALSCHAERT VALVE-GEAR.—This is very generally used on locomotives on the Continent of Europe, though not occupying the predominant position that the Stephenson gear has in American and English practice. A good example is shown in Fig. 391, taken from a German locomotive. We shall refer to this drawing later: but as the basis of an investigation into the movement of the valve, will use the skeleton outline in Fig. 392: here the figure is shortened up somewhat, and the suspension-rig is rather more symmetrically arranged.

The movement given to the valve-slide 8 is the resultant of two components: the first is derived from the eccentric E through the oscillating link 3, and varies in amount as B is moved out from P, and in direction relative to the crank as B is above or below P; the second component is derived from the cross-head through the lever 6. The resultant effect is equivalent to the motion that would be given by a single eccentric, shifting along



is just what would be given by an eccentric diametrically opposite to the crank. Finally, the displacement  $HH_0$  is the same  $BB_0$ , and is increased at  $LL_1$  in the ratio of  $LK$  to  $HK$ : and the total valve-travel  $LL_0$  is the sum of  $L_1L_0$  and  $LL_1$ .

In III. we show how these displacements can be expressed as if directly due to eccentric-radii. To get definite relations, we let

$$\begin{array}{ll} R = \text{crank-arm } OC, & r = \text{eccentric-radius } OE, \\ DP = k_0, & BP = k, \end{array}$$

$$\begin{array}{ll} \text{Maximum length of PB upward} & = k_1, \\ \text{Maximum length of PB downward} & = k_2, \end{array}$$

$$\frac{LH}{KH} = m, \quad \frac{LK}{HK} = n.$$

Then the displacement  $L_1L_0$  would be produced, as stated above, by an eccentric opposite to the crank, and with the radius  $= mR$ , which is shown as  $OE_0$ , with  $OD$  as its travel-distance, the same thing as  $L_1L_0$ . The actual eccentric is laid off at  $OE'$ ; its effective length varies and reverses as  $B$  moves in the link: and in order to combine it with  $OE_0$ , we measure it up or down from  $E_0$  on a line perpendicular to  $OC$ . Thus for the mechanism is drawn.

$$EE_0 = rn \frac{k}{k_0}; \quad . \quad . \quad . \quad . \quad . \quad . \quad (316)$$

While for the extremes we should have

$$E_1E_0 = rn \frac{k_1}{k_0}; \quad E_2E_0 = rn \frac{k_2}{k_0}. \quad . \quad . \quad . \quad . \quad (317)$$

To make (316) strictly general, we should give  $k$  an algebraic sign, indicating the direction of  $PB$ : but it is just as easy to keep track of direction on the figure, and use (316) or (317) as simply quantitative formula.

The geometry of this figure, with the projection  $EF$  of the eccentric  $EE_0$  representing  $LL_1$ , and combining with  $FS$  or  $DO$  from  $E_0O$  to give  $ES$  as the total valve-travel, is very simple. In this

The Joy Valve-gear is a mechanism used in steam engines for controlling the valves. It consists of a horizontal connecting rod, a vertical block, and a valve gear mechanism. The block slides up and down, and its movement is converted into the opening and closing of the valves. The diagram shows the block in two positions, one for admission and one for exhaust. The valve gear is connected to the block and the valves, and its movement is controlled by the block. The diagram is a technical drawing showing the mechanical details of the Joy Valve-gear.

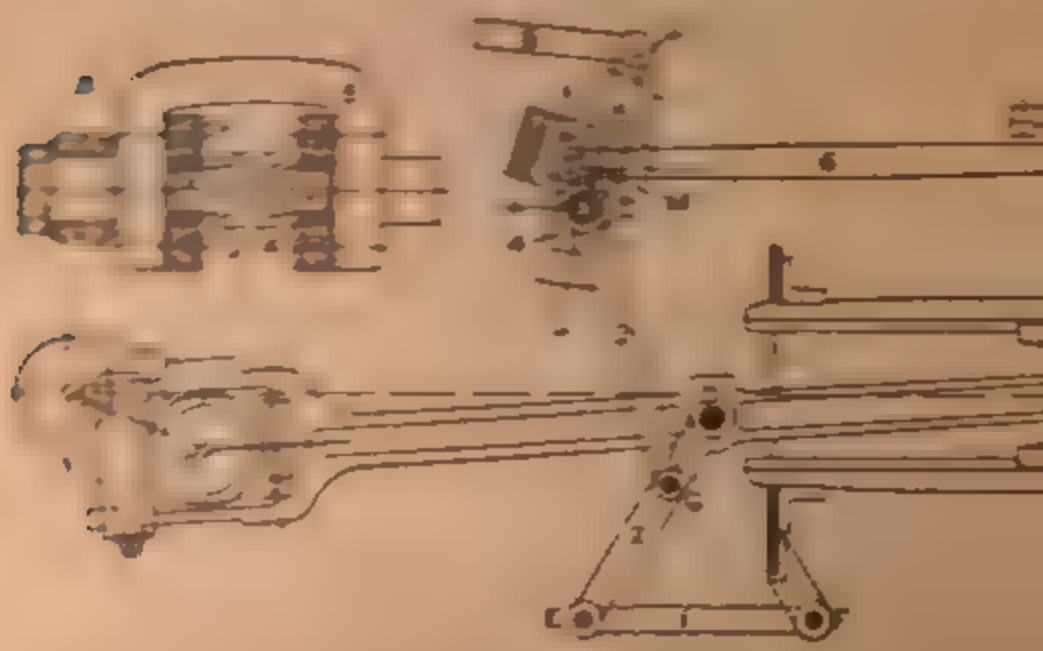
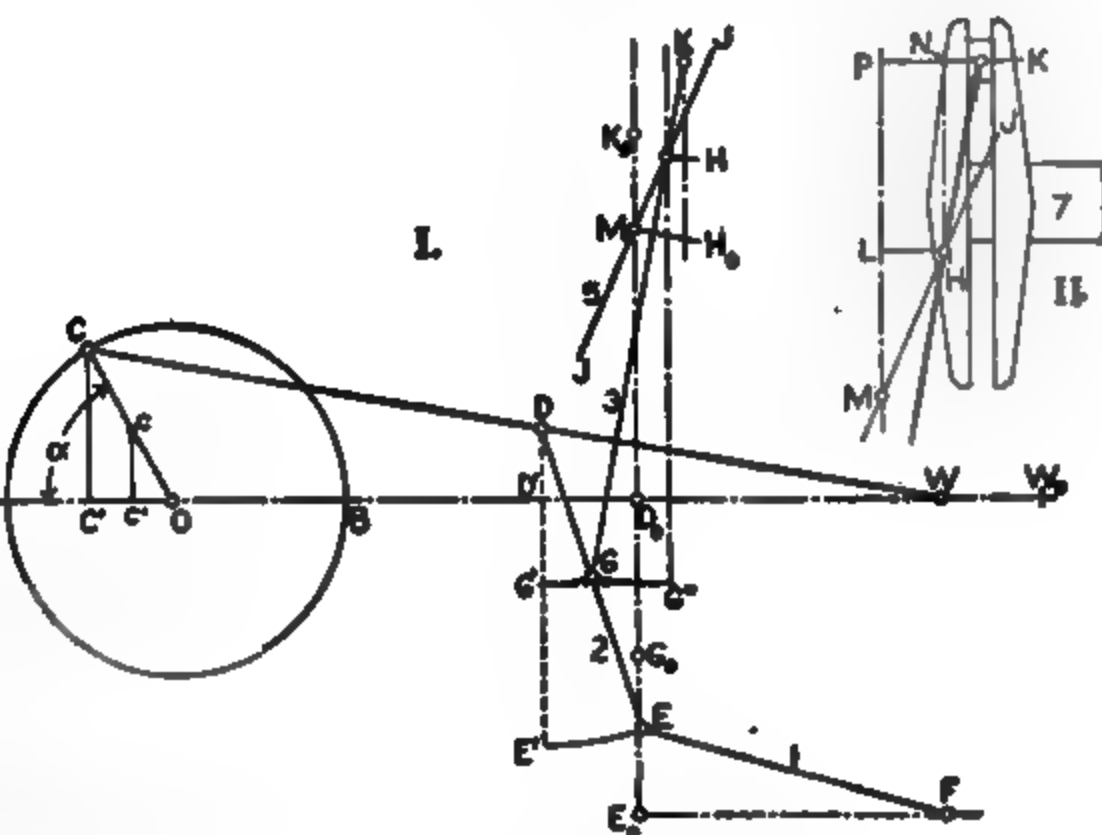


FIG 324 The Joy Valve-gear.

The whole movement is derived from a point near the end of the connecting-rod. The block slides up and down.

ed guide 5: and while this action is essentially different from of the block and link in any of the preceding gears, where was only incidental, we shall nevertheless call this piece 5 link. By changing the inclination of this link, the steam-ribution is changed; and by reversing its inclination from the ical, the engine is reversed. The fact that the valve is driven though by an equivalent eccentric, shifting along a straight, while it is less apparent than with the Walschaert gear, can be established without great difficulty.

(e) KINEMATIC ANALYSIS.—For this purpose, we again have recourse to a skeleton outline, shown in Fig. 395; where the



**FIG. 295.—Outline of Simplified Joy Gear.**

mechanism is modified in one particular, namely, in that the link is made straight, JJ representing its center-line along which the point H on the lever 3 must travel; and the movement of the valve-slide is then taken from the pin K, not through a rod, but by the block and cross-slot shown at II. The deduction is as follows:



the movement NK, due to this swing, will be proportional to the ratio KH/HG, or  $k$ ; and for NK or  $h_2$  we get

$$h_2 = kR(m \cos \alpha + np \sin \alpha). \quad (322)$$

Finally the valve-travel PK or  $t$  will be

$$t = h_2 + h_1 = kmR \cos \alpha + (1+k)npR \sin \alpha; \quad (323)$$

where  $R$  = radius of crank,  $k = HK + HG$ ,  
 $m = GE \div DE$ ,  $p = \tan \text{JMP}$ .  
 $n = DW \div CW$ ,

(f) THE EQUIVALENT SINGLE ECCENTRIC.—Equation (323) is embodied in Fig. 396, similar to Fig. 393 III. Making  $OE_0 = kmR$  (constant), and  $E_0E = (1+k)npR$  (variable with  $p$ ), we see at once that the two parts of  $t$  are given by SF and FE. The limits of  $E$ , at  $E_1$  and  $E_2$ , are determined by the extreme values of the tangent or slant-ratio  $p$ .

(g) THE CURVED GUIDE-LINK.—We must now substitute for the straight guide JJ the actual curved link, in order that the travel may be properly transmitted to the valve as a simple rod. In Fig. 397 I. this

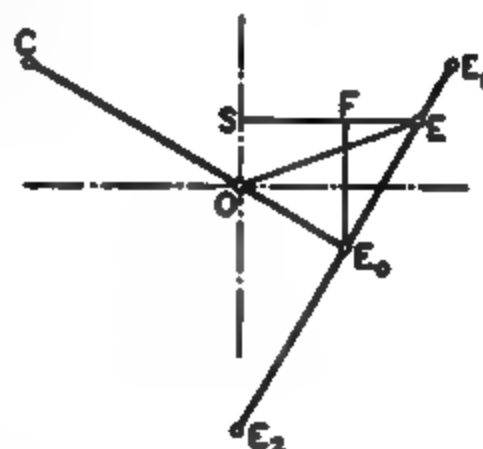


FIG. 396.—Equivalent Eccentric.

part of the mechanism is shown in mid-gear—that is, for  $p=0$ —and also as if D were brought to  $D_0$  in Fig. 395. The link-arc  $J_0J_1$  is struck with a radius HO equal to the length of the valve-rod. The effect of the curved guide is shown by the comparison between II. and III. The former is the same as Fig. 395 II.; in the latter, giving JJ the same inclination from  $J_0J_1$ , we see that PK is just a little larger than with a straight link. The arc  $K_0P$ , struck as if from V in I., is the reference-line for mid-position of valve.

The first of the two main parts of the gear is the valve-chest, which is a box in which the valve is placed. The second part is the link, which is a bar that connects the valve-chest to the eccentric. The link is pivoted at one end to the valve-chest and at the other end to the eccentric. The link is so arranged that it can move in a vertical direction, and this movement is what gives the valve its motion.

The third part of the gear is the eccentric, which is a bar that is pivoted to the crank of the engine. The eccentric is so arranged that it can move in a circular direction, and this movement is what gives the link its motion.

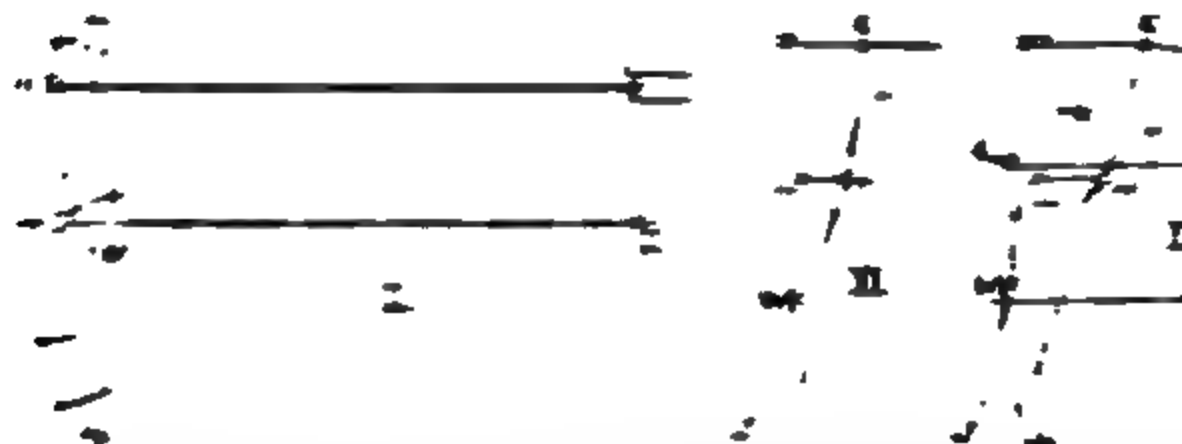


FIG. 24.—Link and Eccentric.

The fourth part of the gear is the valve, which is a piece of metal that fits into the valve-chest. The valve is so arranged that it can move in a horizontal direction, and this movement is what gives the valve its motion. The valve is pivoted to the link, and the link is pivoted to the eccentric. The valve is so arranged that it can move in a horizontal direction, and this movement is what gives the valve its motion.

**Advantages of the Rodless Gear.**—These valve-gears have the advantage that they require very little room sideways. The motion is in a direction measured along the axis of the engine-shaft, so that where the engine is cramped for space, as in a locomotive with inside cylinders, they can be more easily accommodated than the Stephenson gear. And when the valve-chest is most conveniently placed on top of the cylinder, as in the locomotive, or in



if it in a marine engine—that is, in general, in the plane main working mechanism—then these gears obviate the need of a rock-shaft to transfer the movement. They have more than the link-motion, unless the latter is complicated by an extra transmitting-rig, as in Fig. 366: and in the Joy gear, the curved guide is not a very good feature from the point of view of lubrication and wear.

They are used chiefly on locomotives, sometimes on marine engines. The locomotive from which Fig. 394 was taken is a cylinder compound; the inside cylinders have the Joy gear illustrated, the outside cylinders have Walschaert gears. In the latter, the return-crank is a very effective device for carrying the eccentric.

### § 53. The Double-valve Gear.

(a) GENERAL CONSIDERATIONS.—It has been made clear by Figs. 338, 348, and 351 that when early cut-off is secured with a slide-valve, the lap must be relatively very large and the port-opening small; and that the compression must begin early in the return-stroke, on account of the great angular advance of the eccentric. To prevent early compression from running too close to the exhaust, the clearance must be large; but while large clearance is essential to the design, and unavoidable, in most high-speed engines, and while a good compression is mechanically advantageous when the inertia-force of the reciprocating parts is large, a closer approach for economy renders desirable a valve-action which will cut away so much of the possible effective area of the steam pressure diagram as is lost, for instance, in Fig. 31. Simpler than the arrangements with separate valves and ports for admission and exhaust, and with no limitations as to speed, the Meyer or double-valve gear is the first type which we shall consider as meeting the requirements of full port-opening, quick cut-off, and the most favorable regulation of the exhaust operations.

(b) THE MEYER VALVE-GEAR.—In Fig. 398 the main valve  $V_m$  is extended beyond the lap-edges, so as to enclose the ports P, P;



This crank-arm rotates about a moving point  $E_m$  is immaterial; but this movement of the center of rotation can be eliminated by shifting  $r_v$  to  $OE_v$ , or to the opposite side of a parallelogram of eccentrics: then since  $SE_v$  is always the same as  $SE_m$ , we have that  $V_s$  would be driven by a fixed valve-seat by  $OE_v$  as it is actually driven on a moving seat by  $E_mE_s$ . The virtual eccentric  $r_v$  we call the virtual eccentric; and its angle  $\delta$  is measured as indicated on Fig. 399.

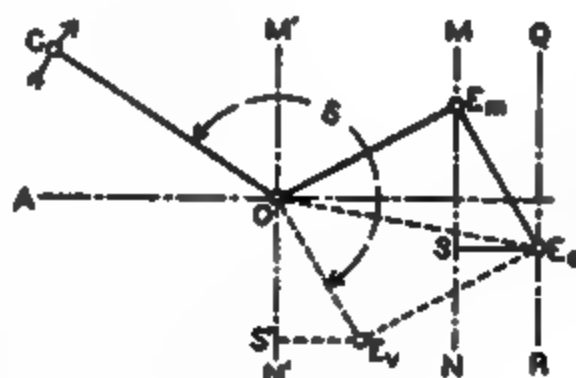


FIG. 399.—Relations of the Eccentrics.

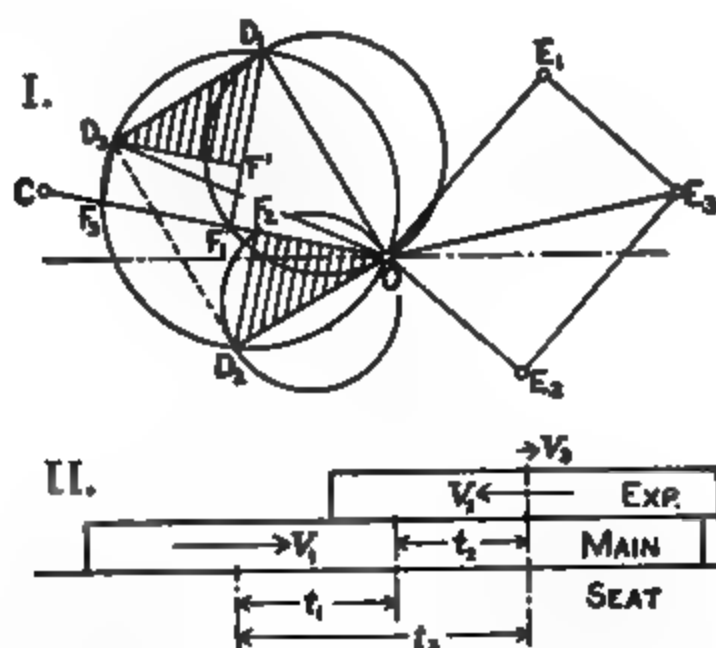


FIG. 400.—Relative Motion Illustrated.

The ideas just stated are expressed in yet more graphic form in Fig. 400, where the three valve-circles are drawn in the usual manner, and the crank is taken at any position  $OC$ . The relation between the three travel-distances, namely,

$$OF_3 = OF_1 + OF_2, \text{ or } t_3 = t_m + t_v, \quad \dots (324)$$

is clearly shown on I.; this is further brought out at II., as also the fact that the absolute velocity  $V_s$  of the riding-valve is the

resultant of the absolute velocity  $V$ , and the relative velocity  $V_r$ . It will be noted that the two valves are never simultaneously in mid-position, as drawn for illustrative purposes in Fig. 398.

(d) FUNCTIONS OF THE CUT-OFF VALVE.—It is required of the valve that, being out of the way when main-valve admission begins, so as to leave a clear passage into the port P, it shall then close this port quickly at some desired instant, earlier or later in the stroke of the piston. Any crank-driven slide moves most rapidly when near the middle of its stroke; consequently, the cut-off valve must have a small lap if it is to be in rapid motion when closing its port. Very often the lap is negative, or the valve does not cover the ports when in mid-position, as is the case in Fig. 398.

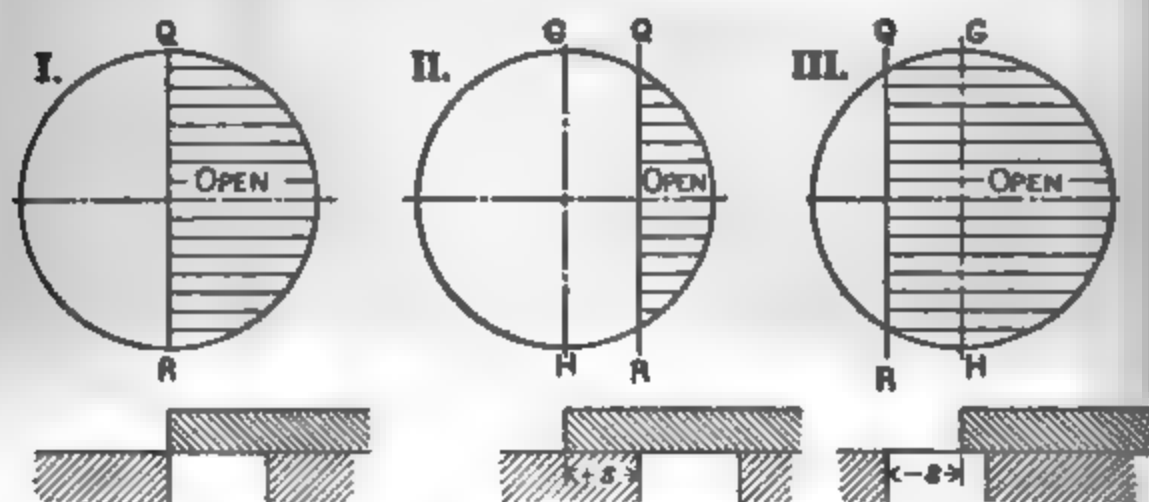


FIG. 401.—Effect of Varying the Lap.

(e) POSITIVE AND NEGATIVE VALVES.—Certain ideas fundamental to the whole matter of valve-action are given concrete and complete expression in Fig. 401. By means of primitive diagrams of the ordinate type (the simple eccentric diagram Figs. 331 and 332), we emphasize the fact that the important result sought in any determination of valve-motion is the movement of the valve-edge with reference to the port-edge, rather than of the center of valve with reference to center of seat—this by markedly substituting the lap-line QR for the center-line G as a reference-line. Further, we show in II. how with positive lap the period of opening is less than half a revolution, so that the single valve is necessarily of this type, on the steam side

while the valve with negative lap, or the negative valve, opens its port for more than half a revolution, and can therefore perform only one operation. Rotating II. into the regular Reuleaux position, we easily see that the eccentric driving a single valve must, of necessity, be in the second quadrant ahead of the crank if the valve is direct, in the fourth if it is indirect.

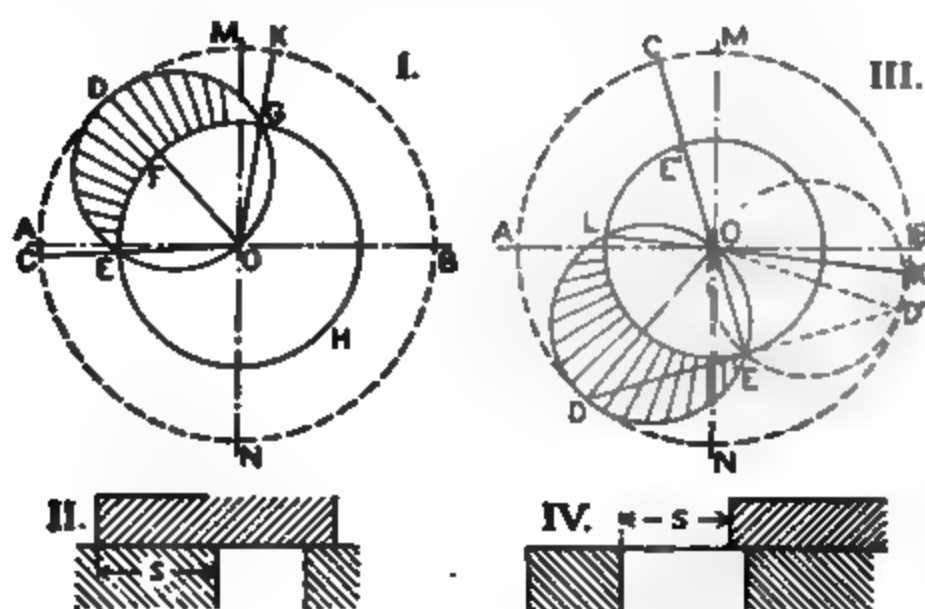


FIG. 402.—Locating the Valve Diagram.

II) ECCENTRIC-SETTING FOR THE NEGATIVE VALVE.—In Fig. 402, Zeuner diagrams are used to illustrate these same ideas, to show how the eccentric-setting for a negative valve is determined. I. and II. are entirely obvious, but we now lay special stress on the facts that the crescent EDG is a diagram of port-opening, and that the cut-off intersection G determines the crank-angle for which the valve is at the distance  $s$  to the right, and is moving toward the left.

With the negative lap  $-s$  shown at IV., let us require that the valve close the port when the crank is at OC in III. The lap-circle is drawn with absolute radius  $s$ , algebraic sign having no immediate significance. For cut-off, the valve must be to the left, wherefore the valve-circle must necessarily go through E instead of E': and the fact that the negative travel-intercept must increase as the crank advances puts the circle in the full-line position on OD, as against that on OD'. The crescent EDL is

now a diagram of port-closure; and we see further the characteristic of a negative cut-off valve, if of the expansion type, to have its valve-circle in the fourth quadrant and its virtual eccentric in the third quadrant ahead of the crank.

(g) VARYING THE CUT-OFF.—The action of the expansion valve is shown by Fig. 403, where cut-offs are determined by quarters from minus three-fourths to plus three-fourths of the radius of the virtual eccentric. Through comparison with Fig. 402 III., the diagram is self-explanatory. On the



FIG. 403.—Control by Variable Lap

especially marked lines, the cut-off is designated +3 and -3. As the valve-edge approaches the crank, it moves more rapidly from the positive to the negative of the valve-circle, as is clearly shown. Another example of a large negative cut-off is shown, where the closure is short, and the valve is very early, the eccentric must reopen its port before the

crank cuts off. This possibility exists only with expansion valves and belongs rather to the method of regulation than to Fig. 404.

In Fig. 404 I., with a negative lap equal to the radius of the valve-circle, the valve-circle is located for cut-off at 0.1, at 0.3, and at 0.5, and at II., is given a diagram showing how the expansion-eccentric will have to be rotated to secure this effect. It is a simple matter thus to locate the eccentric at the center of the first, and place the crank at the center of a shaft-governor of suitable design. A wide range of adjustment would be covered than is here shown; and the eccentric must be rotated by the governor in proportion equal to the crank-angle between earliest and latest cut-off.

(h) THE BUCKLEY VALVE-GEAR.—In Fig. 405 is shown an interesting device whereby the effect of a rotation of the crank about the main eccentric is secured from a governor which rotates the expansion-eccentric about the center of the main

of the compound rocker-arm, at II., helps to make clear action. The cut-off rocker GBD is pivoted at or near the center of the main rocker AF; and the path of the lower pin G

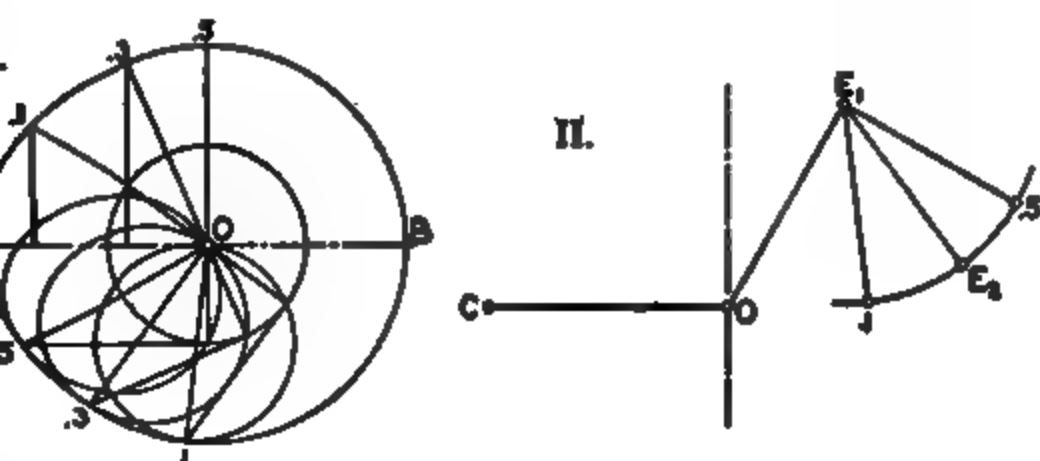


FIG. 404.—Rotating the Virtual Eccentric.

is right by the fixed center A. If  $E_2$  were brought to O and fast there, the two rockers would swing together almost as a piece: but if G receives from  $E_2$  harmonic motion with respect to O—or, to be more precise, with respect to its own mid-position

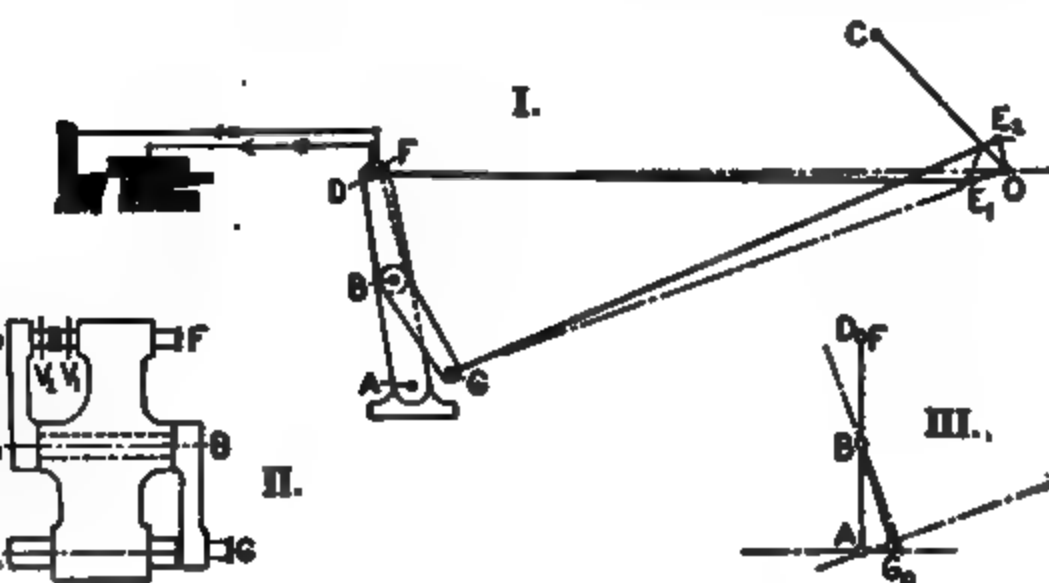


FIG. 405.—The Buckeye Valve-gear.

shown at  $G_0$  in III.—this absolute motion will be changed to relative motion of D with respect to F at the top of the rockers. The reversing effect of the riding rocker-arm is neutralized in the same manner, by reversing the eccentric. A partial section of

the valves, taken from Fig. 253, shows their relation with the main port not quite fully open, and the steam getting ready to close. Both eccentrics are in the position,  $E_1$ , because its valve is of that type.  $E_2$  is at another position for rocker reversal.

At III. the two rocker-arms are shown in motion. Approximately,  $BG_0$  is at right-angles with the line  $AO$ . It appears that the effect of secondary movements of  $G_0$  is small if  $G_0$  is a little to the left of the foot of the perpendicular upon  $AO$ . This is, however, a matter which would require a detailed investigation, besides being of small relative importance. Of course, the angle which the stroke-line of the expansive eccentric  $E_2$  makes with that of the engine must be taken into account in setting this eccentric.

(i) VARYING THE VIRTUAL ECCENTRIC.—Moving the center of this eccentric along an arc struck from the main eccentric is:

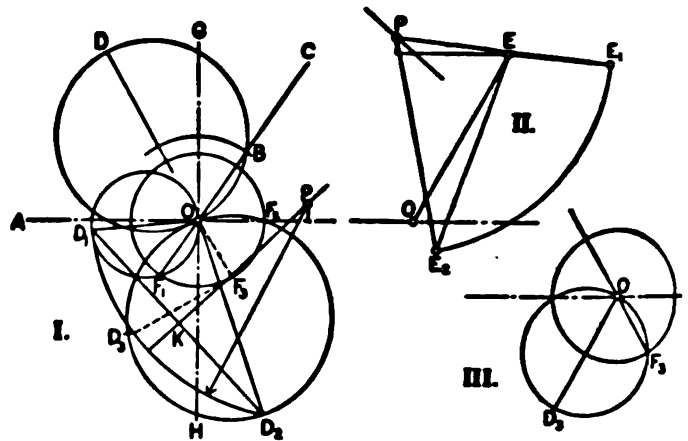


FIG. 406.—General Case of Change in Virtual Eccentric.

Fig. 404, is only a particular case of the general operation of shifting this center along a definite locus. The problem illustrated in Fig. 406 is more general in its terms, and serves to bring out some important properties of the double-valve gear. Having



main valve-circle OD and the lap OB, we find the latest cut-off C: and the problem is, to design a gear which will change cut-off all the way from OC to zero, or to OA, with the least linear movement of the virtual eccentric, but with variation in length of this eccentric permitted.

Referring to Fig. 402 III., we see that, with a given negative shortening the eccentric-radius will make the cut-off later: since to turn OD toward ON, against the direction of crank-revolution, is equivalent to advancing the eccentric, the smaller eccentric will not have to be turned so far back for a late cut-off as a larger one. In Fig. 406, for the small circle OD<sub>1</sub>, this fact is shown by the great convexity of the segment based OF<sub>1</sub>, as compared with that on OF<sub>2</sub>: which is another way of saying that angle F<sub>1</sub>OD<sub>1</sub> is less than F<sub>2</sub>OD<sub>2</sub>. These considerations make it clear that, with a determinate angle F<sub>1</sub>OF<sub>2</sub> of cut-off variation, the angle D<sub>1</sub>OD<sub>2</sub> of eccentric variation is made less by using a small valve-circle for the late cut-off and a large one for the early cut-off.

Having chosen D<sub>1</sub> and D<sub>2</sub>, the natural locus connecting them is a circular arc from some point on the perpendicular KP, drawn from the middle of the chord, D<sub>1</sub>D<sub>2</sub>. The farther away P is taken, the less will be the angular movement of the swinging eccentric. Fixing upon a particular position of P, we draw the locus of D. Then transferring O to E in II., and reversing the whole figure with reference to a vertical axis, we get the locus E<sub>1</sub>E<sub>2</sub> of the expansion eccentric, on the plane of the shaft-governor. With early cut-off, it is interesting to note how small becomes the radius of the expansion-eccentric, which gives the absolute motion of the riding-valve.

At III. is drawn the valve-circle for quarter cut-off, the dotted line in I. showing how D<sub>2</sub> is located. It does not appear that considerations as to steam-distribution will have much influence in establishing a certain curvature of D<sub>1</sub>D<sub>2</sub>D<sub>3</sub> as the most desirable.

The advantage of the regulation in Figs. 404 and 405 is, that the cut-off valve has a constant travel on the main valve, so that wear will be uniform. In the case of a valve with variable travel, but with a stroke less than the maximum prevailing most of the

time, there will be a non-uniform wear of the seat, leaving a slight shoulder or rise at the limit of the usual travel.

(j) CUT-OFF VALVE WITH THE STEPHENSON LINK-MOTION.—This is the last case of the application of the riding-valve which we shall investigate, using the diagrams given in Fig. 407. At I. is shown the arrangement of the eccentrics, the cut-off valve being driven by the fixed center  $F$ , so that the virtual eccentric

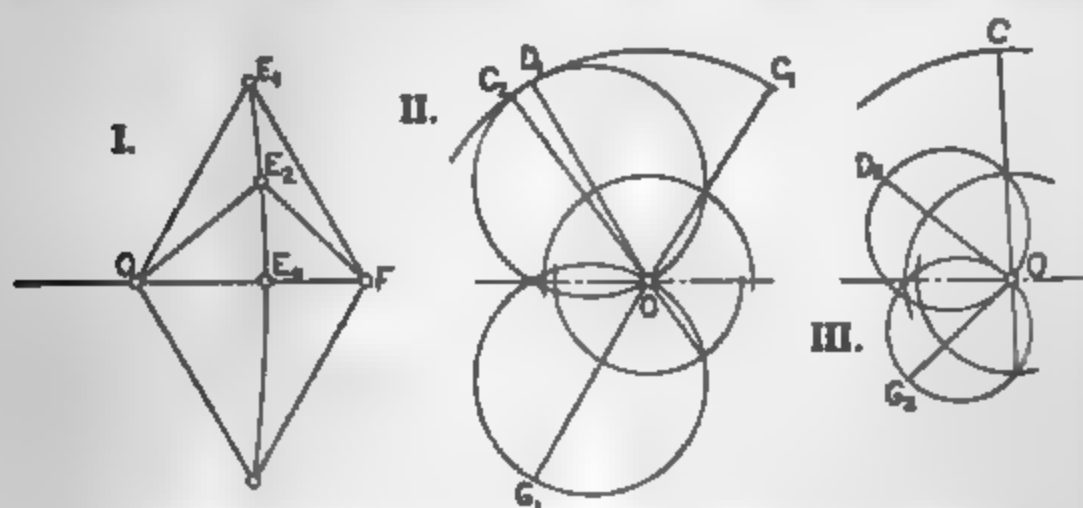
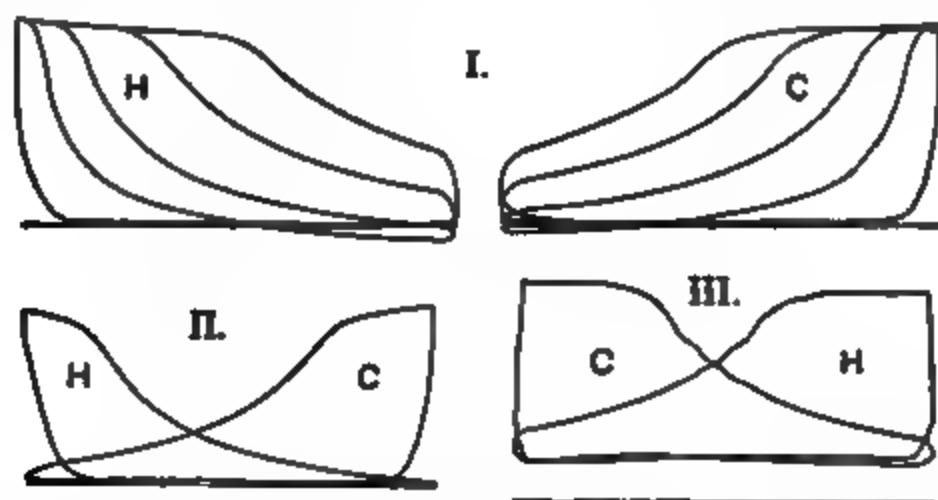


FIG. 407.—Diagrams for Cut-off Valve with Link-motion.

is  $E_1F$ ,  $E_2F$ , etc. Valve-diagrams for full gear forward are drawn at II., showing main-valve cut-off at  $OC_1$ , that by the second valve much earlier, at  $OC_2$ . The proportions of the figure are so chosen that for half-gear, at III., the two cut-offs are simultaneous, and at the latest possible point. It appears that this arrangement would be suitable for a reversing-engine which was to work with a high ratio of expansion; but it gives a rather peculiar manner of variation of cut-off with reverse-lever position. It has been used on compressed-air motors.

(k) INDICATOR DIAGRAMS from engines with the double-valve gear are given in Fig. 408. The first set was taken from an engine with a load that fluctuated continually and widely, the effort to follow it keeping the governor dancing. The several expansion-curves traced are selected from a large number of closely-spaced lines: constant compression and a quite sharp cut-off are the distinguishing features of these diagrams. At II. the admission pressure drops as the piston speeds up toward mid-stroke, prob

because the ports in this rather old engine are inadequate in but the cut-off curve is still quite short. Rather different arms, with late admission and very short compression, are



408.—Indicator Diagrams from Engines with the Double-valve Gear. I. Buckeye Engine, 16"×24", 145 R.P.M.; II. Buckeye Engine, 16"×22"–110; III. Air-compressor, 10"×12"–145.

at III. Compare with all these the Corliss-engine diagrams Figs. 465 and 476.

## § 54. Various Forms of the Slide-valve.

(a) TYPES OF VALVES.—The simplest possible plain slide-valve that used for illustration in Figs. 9 and 334: it makes provision for the essential functions of the steam-distribution. In practical application this primary valve is modified and complicated along two lines: usually the valve is "balanced," or arranged so that it will not be forced hard against its seat by the steam-pressure; very often the valve is so formed that it will admit flow past several edges, or it is made "multiple-ported." A review of the cylinder drawings given in § 43 will show pretty clearly the following three typical forms of the slide-valve:

1 The single-faced flat valve with a relief device on the back, as in Figs. 247 to 249.

2 The double-faced flat valve, usually under a rigid balance-seat, as in Figs. 6 and 7 and Fig. 250. The double-seated valve in Fig. 251 is a special form under this heading.

### 3. The piston-valve.

In illustrating and describing a large number of valves beside those already given, we shall use two principal points of view in examining the valves with regard to—

A. FORM, with reference to the functions of the steam-distribution;

B. BALANCE AND TIGHTNESS, and incidentally the possibility of relieving excessive pressure in the cylinder.

(b) LOCOMOTIVE VALVES, when flat, are usually of the first type; one good example has been shown in Figs. 247 to 249 and another is given Fig. 409. The balance-rig is quite different in form in the two cases, though the same in effect; and it is a distinctive feature of both designs that this rigging is flexible, permitting the valve to rise easily for relief of the cylinder. Further, this flexibility obviates the need of extremely accurate parallelism between the face of the balance-plate and the valve-seat—a condition which is rather difficult of perfect attainment with the copper-packed joints between the cylinder-table and the valve-chest cover.

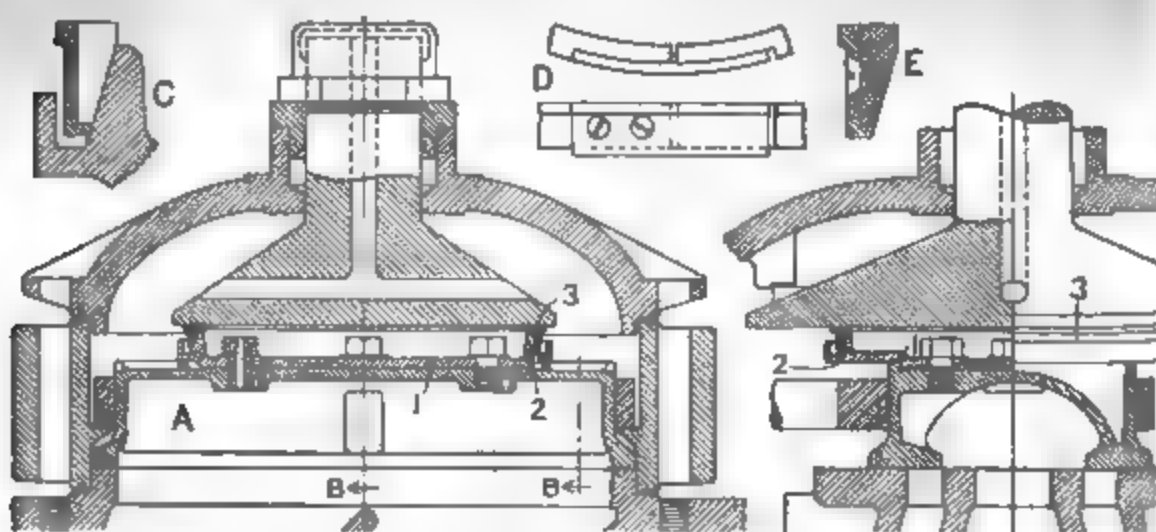


FIG. 409.—Locomotive Valve with "American" Balance-rig; special design by the Pennsylvania Railroad. Scale 1 to 12.

In Fig. 409, the circular plate 1, securely bolted upon the back of the valve, carries a cone 2 on which fits the spring-ring 3. The ring is cut and has a joint formed as shown in detail at C, D, and E. In the normal position of the valve, the ring is sprung off

little, so that its own elasticity keeps it up against the balance-plate: in action, it is forced tightly against both plate and by the pressure of the steam. If the valve is lifted from seat, the ring opens; but slides up again on the cone as the valve returns. The space inside the ring is relieved to the exhaust through the hollow tap-bolts. Valves made under this system sometimes have the cone formed right upon the main casting instead of on a separate plate—especially when two small valves are used instead of a single large one.

The balance-plate in this particular design is of unique and unusual form—being supported by a big central stud cast with it, and drawn up by a single large nut against four lugs, as shown on the left side of view B, which steady it against any tendency to rock: in the center of the stud is an oil-conduit, branching to the sides of the plate. The type of plate in Figs. 247 and 248 is, however, decidedly the usual thing.

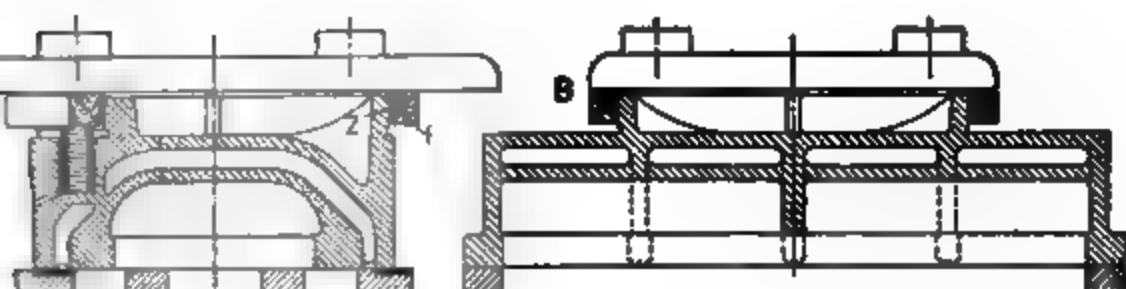


FIG. 410.—German Locomotive Valve with Allen port. Scale 1 to 15.

Fig. 410 shows another valve with a circular relief-ring: here the outer ring 1 is made solid and held up by four helical springs in pockets; the inner ring 2 is cut, and its cylindrical inside surface fits against a segment of a sphere formed upon the valve. A special feature of this valve is the auxiliary port, which passes steam over from the far side of the valve to supplement the regular admission; the two equal steam-laps which control admission at the left-end port are emphasized in the figure, that for direct admission at the left, the other at the right end of the valve. Writers in English call this the Allen valve; in German it is the Dick valve, from a contemporaneous inventor. When a single valve is to give early cut-off, it is entirely logical thus to provide

le admission on the steam side with only a single op  
must, because of the great inequality in the respo  
in the widths of opening of the port. In this figure  
ded negative lap, or "clearance," on the exhaust side

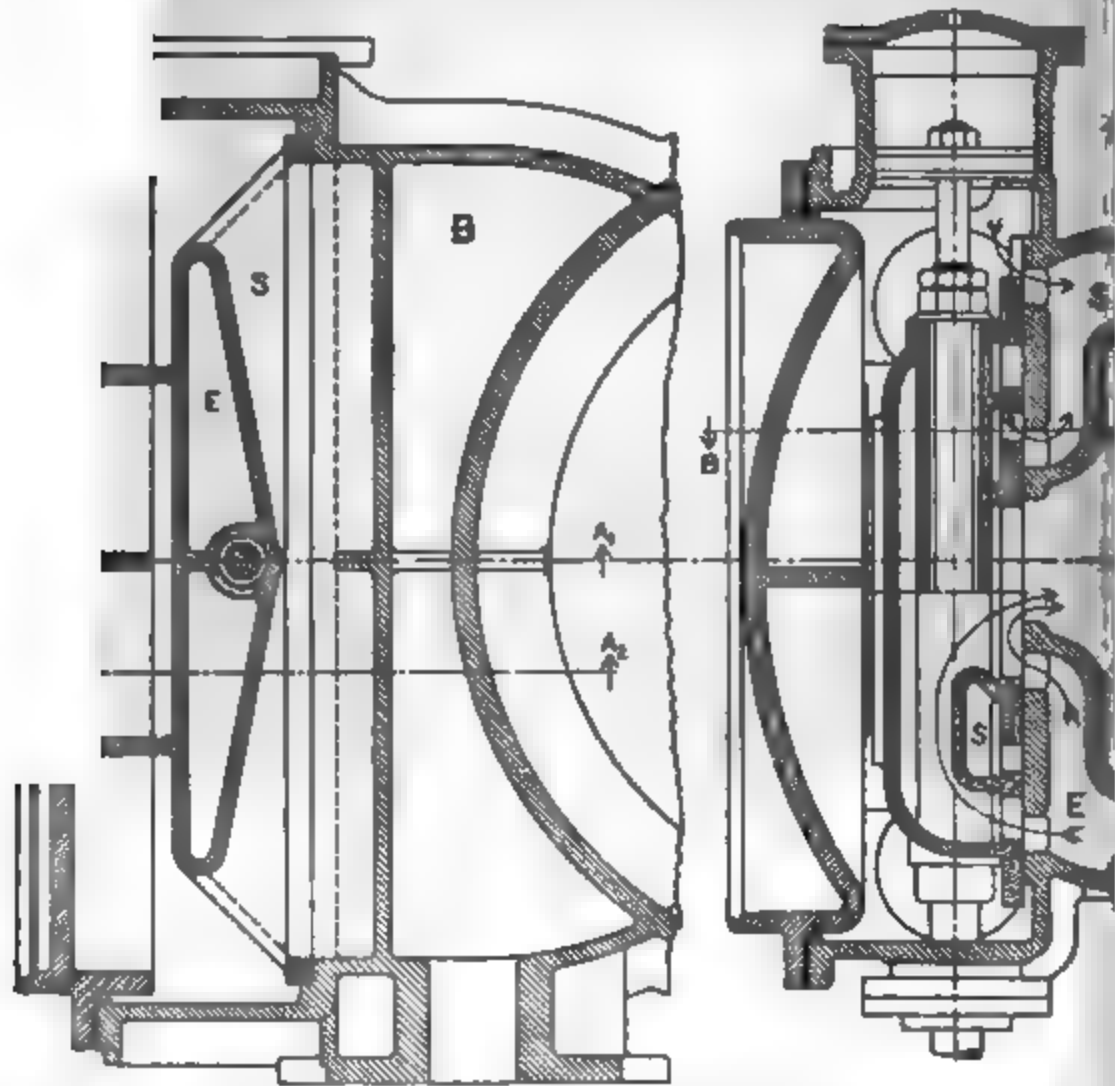


FIG 411 —Double-ported Valve for low-pressure cylinder of engine  
shown in Figs 259 and 260 Scale 1 to 12

(c) FLAT VALVES FOR MARINE ENGINES.—Flat valves  
been used in some rare cases for the higher cylinders of m  
engines; but on the lower-pressure cylinders they are very  
mon. Double opening is generally secured by the arrange  
shown in Fig 411. Inside of a long D valve, which has a wot  
face to control each of the outer ports, is placed a second v  
face with the same laps, for each inner port; and there mu  
room to form over and outside of this inner face a pocket or c

ber which will conduct steam inward from the sides of the valve-body. The action as to steam-flow is clearly shown by the arrows in view A, and the form of the steam-pockets by the section of the valve at B. This rather small valve has no relief-device, but there are guides on the valve and on the chest-cover to keep the valve from falling away from its seat when steam is shut off. To permit rising for cylinder-relief, the hole for the valve-rod is made with considerable clearance, and the nuts are not screwed down tight enough to grip the valve. The use of a lifting or compensating piston, to balance the weight of the valve and perhaps a part of the weight of the valve-gear, and thus equalize the pressures on the eccentrics, is almost universal. On the higher valve-chests, the top of each balance-cylinder is connected to the next receiver; on the low-pressure chest the pipe runs to the condenser.

A larger valve of this same double-ported type is shown in Fig. 412. As to the valve itself, we note the lightness of the casting and the free use of stiffening-ribs. Other notable features are, the "false" or separate valve-seat, fastened to the cylinder-casting by sunk-head screws, made of hard bronze, and renewable when worn out; and the special form of relief-ring, connected to the body of the valve by an annular diaphragm made of copper or of some copper alloy: the detail of this device is shown at D.

Sometimes the auxiliary valve-face of Figs. 411 and 412 is duplicated, giving triple opening, as in Fig. 413. With this increase in effectiveness goes an increase in the size of the valve and its greater complexity as a casting, together with a larger clearance-volume in the cylinder. There are even cases on record of four-ported valves of this type, but that seems to be going too far. In the figure under discussion we have a type of relief-ring which is more in accord with regular marine practice, in that it is held in the cover and slides on the valve, instead of moving with the valve as in previous examples. Further, the ring is circular, instead of being rectangular as in Fig. 412 and in earlier designs which somewhat resembled Fig. 249. But in its detail, shown at B, this relief-ring is rather peculiar, the type of arrangement shown in Fig. 423 being usual. Here the fitting-ring 2

appears to be pressed into the cover and fastened with screws; between 2 and 3 there is a close sliding fit; and 3 is pressed against the valve 4 by a circular spring marked 5.

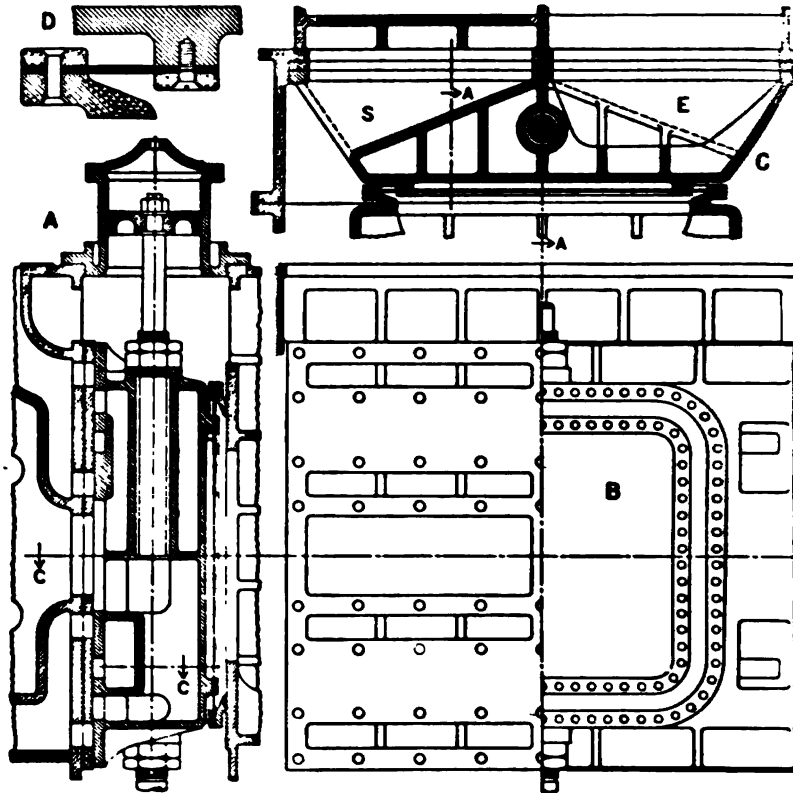


FIG. 412.—Valve for Low-pressure Cylinder of engine partly shown in Fig. 273 I. Scale 1 to 24 and 1 to 6.

Occasionally, with large valves of this general type, the exhaust steam is carried out through an opening in the back of the valve, inside the relief-ring, instead of going into a passage formed under the valve-seat; then the seat can be brought closer to the cylinder.

Fig. 414 shows an interesting adaptation of the two-faced flat valve under a rigid balance-plate, with double opening for both





mission and exhaust. The form of the plate, with the manner in which it is supported and held in place, is clear from the draw-

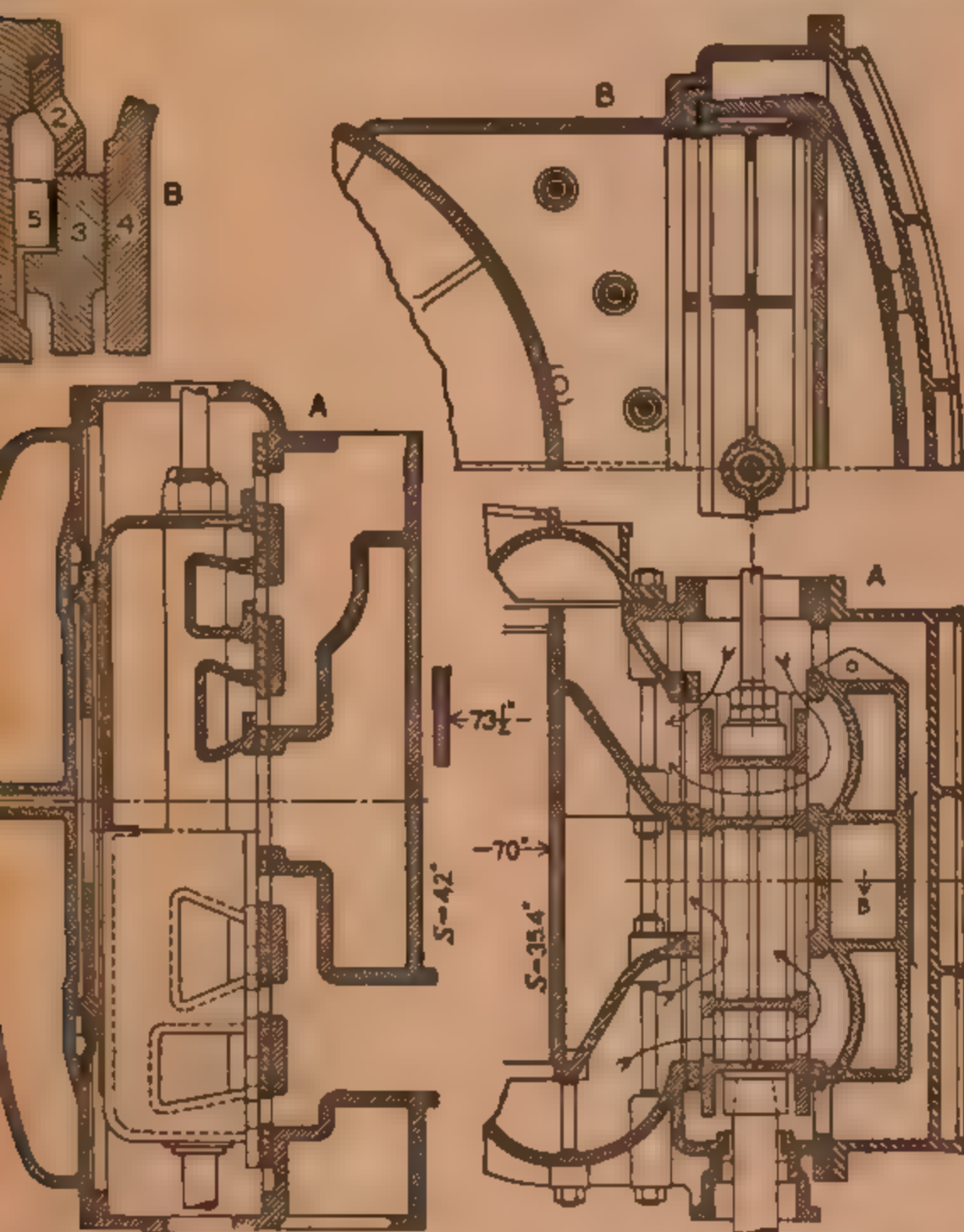


FIG. 413.—Triple-ported Valve, recent British cruiser Scale 1 to 24.

FIG. 414.—Large Two-faced Valve, for L. P. cylinder of cruiser engine. Vulcan Works, Germany. Scale 1 to 24.

Fig. As a structural detail, we note the free use of stay-bolts to brace the port-walls, already referred to in § 43 (r).

(d) SPECIAL FORMS OF THE FLAT VALVE.—A possible though seldom used form is the B valve, sketched in Fig. 415. In this,

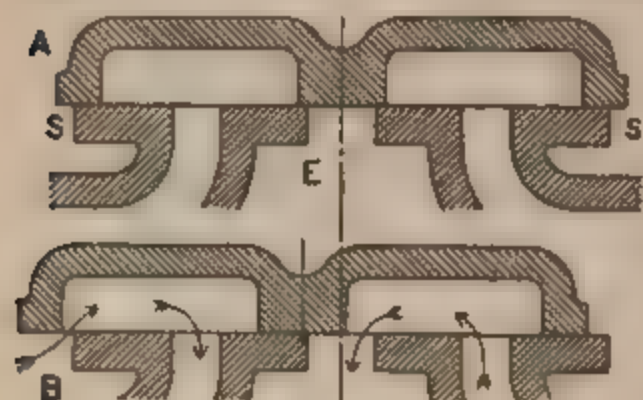


FIG. 415.—The B Valve.

admission as well as exhaust takes place through a cavity in the under side of the valve, and with steam filling the chest as with the ordinary direct valve, the movement for opening and closing is that of the indirect valve. For a given set of proportions—port-widths and laps

—this valve necessarily exposes a larger area to the steam-pressure than does a D valve. In the form here shown it is found only in some small simplex pumps; but its essential form and action are frequently embodied in the piston-valve, as for instance in Figs. 254, 255, 436, and 441.

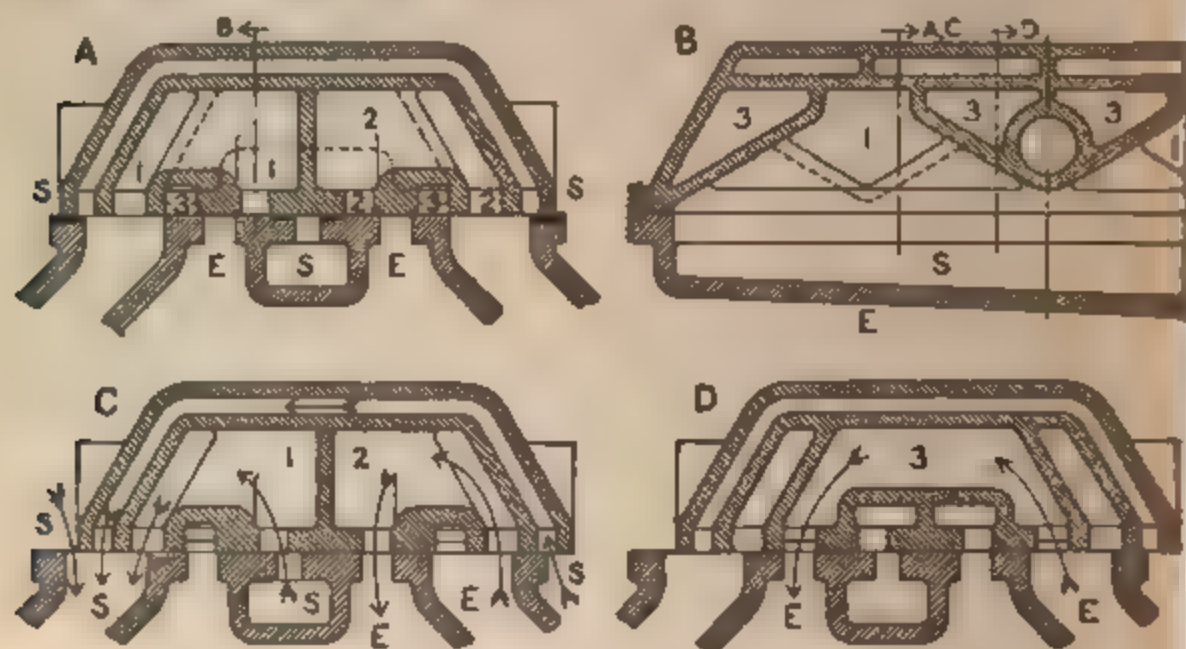


FIG. 416.—A Very Complicated Flat Valve.

A single-faced flat valve with triple opening for admission and double for exhaust is shown in Fig. 416. It is of German design and appears to be at about the limit of possible or profitable complication. The special feature of the valve-seat is the steam pocket and extra port in the middle, the pocket opening at on

nd into the steam-chest. The valve has first an Allen port of the usual type; inside of this are the chambers 1 and 2, which are separated by a middle partition, and from which steam can pass from the middle port to either main port; and besides these there is another chamber 3, analogous to the Allen port and best shown in view D, which furnishes the second passage for the exhaust. How these various passages combine without interference can be seen in the cross-section at B. The several laps are blacked on view A, those for admission upon the valve-seat, the exhaust-laps upon the valve.

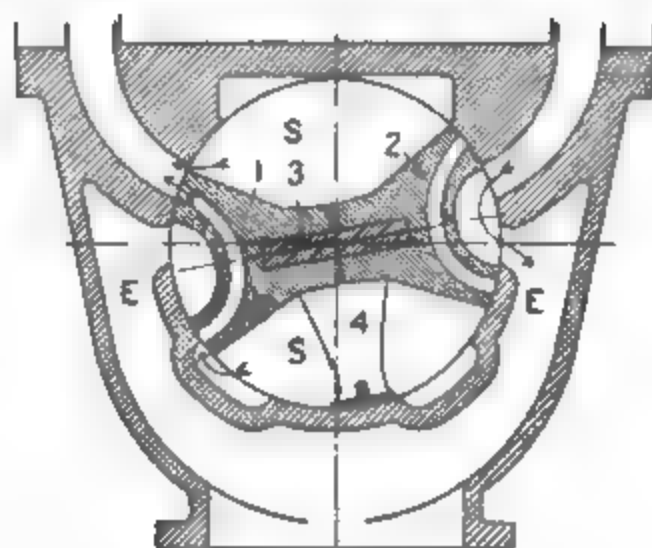


FIG. 417. — Oscillating Valve for Low-pressure Cylinder of tandem-compound engine.

Occasionally a valve with the essential form of the common slide-valve is made to oscillate on a curved (cylindrical) seat, somewhat after the manner of a Corliss valve. An example much more complicated than a simple adaptation of the D valve is given in Fig. 417: there are really two valves, 1 and 2, one for either end of the cylinder, each with an Allen port to give double admission. These are moved by the spindle 3, which is broadened in the middle to a flat slab; and the weight of the whole valve is carried by the support 4, which is fast to the spindle. The latter passes out through a stuffing-box and is oscillated by a crank keyed to it and driven by the eccentric-rod.

(e) PROPORTIONING THE VALVE.—Suppose that, having decided upon the width of the cylinder-ports and the laps, we wish to design a slide-valve which shall be as short as possible, with a minimum area subject to steam-pressure. The determining condition is illustrated in Fig. 418 I., where the valve of Fig. 247 is shown at its greatest distance from mid-position: evidently the width of the exhaust-port and of the port-walls or "bridges" must be such that the least opening between the edges A and D shall be not less than the width of the port C. Similarly, the valve of

Fig. 415 is here shown at its limit of travel in II., because any further movement toward the left would choke the exhaust from the right-end port.

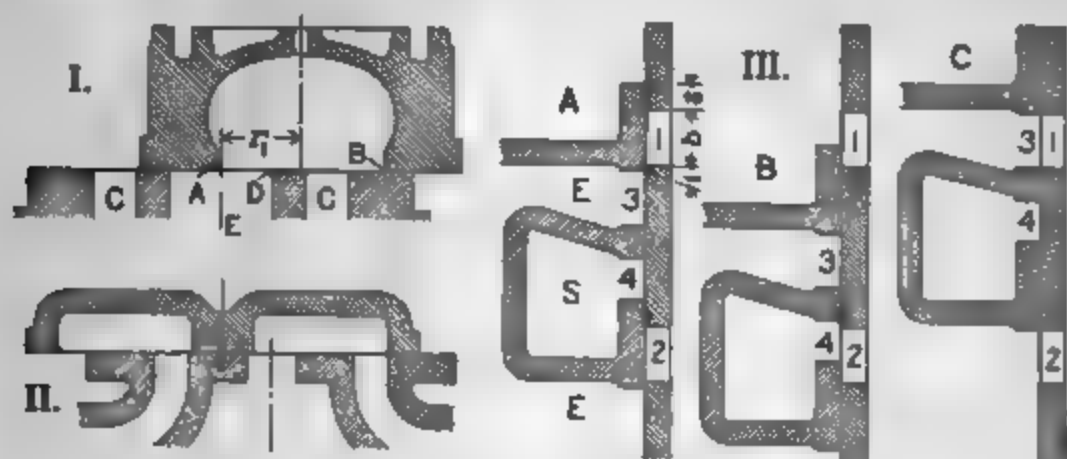


FIG. 418.—The Critical Position of Several Valves.

To build up the face and seat of a valve of complex form, with due regard to this requirement of non-interference of function, is an operation calling for an intelligent use of the cut-and-try method. As further illustrating the character of the relations involved, consider the example in Fig. 418 III., which is a detail from Fig. 413, with A showing mid-position and B and C the two extremes of travel. The dimensions  $b$ ,  $s$ , and  $i$  are fundamental. At C the valve gives full opening for exhaust, and its proper limit of travel

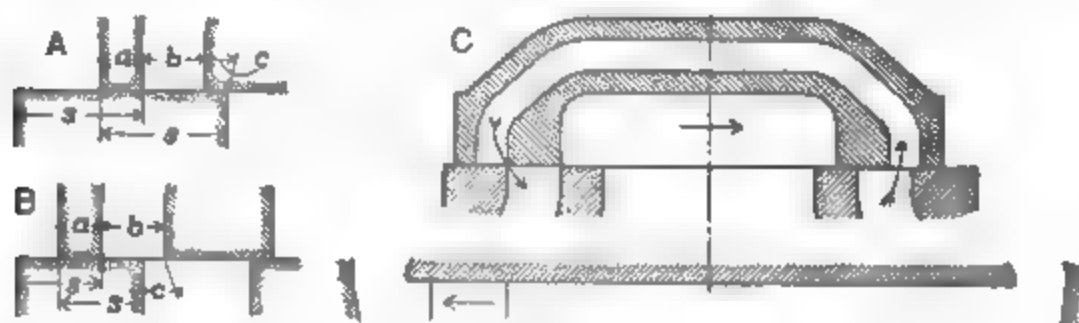


FIG. 419.—Design of the Allen Valve.

is here determined, unless opening 3 can be made wider than port 1; at the least, 3 should be equal to 1. With the same displacement in the other direction, at B, the greatest opening for admission is less than the port-width by the amount  $(s-i)$ , since in II.  $r = (b+i)$ ; and it is evident that the opening 4, as here drawn, has a little greater width than can be made effectively useful. This

ing, with the width of the wall-face between 3 and 4, fixes the distance between the ports 1 and 2.

*Negative Lap with the Allen Port.*—When a valve of the Allen type has a large steam-lap, it is not difficult to keep the opening of the auxiliary port within this lap, and even to leave a small positive lap  $c$  at its inner side, as in Fig. 419 A and Fig. 410. But when the main lap  $s$  is smaller or the width  $b$  is increased, there will be a negative lap to the Allen port, as at  $c$  in Fig. 419 B and Fig. 416. The result is, that during a brief time a passage is opened from one end of the cylinder to the other, with some tendency to modify the exhaust operations. Thus at C in Fig. 419 we have the valve yet at the distance  $c$  to the left and moving toward the right: compression has begun in the left end of the cylinder, and now steam which has not yet gotten to release in the right end is about to flow over and increase the pressure and the amount of the clearance-steam in the other end. This action will persist while the valve travels through the distance  $2c$ ; but since it moves rapidly near mid-position and the openings involved are small, the effect upon the steam-distribution will be relatively insignificant.



FIG. 420.—Unbalanced Pressure.

(f) *BALANCING THE FLAT VALVE.*—In order to develop certain ideas which must underlie this discussion, let us consider the simple arrangements outlined in Fig. 420. Case A, where a flat plate merely rests upon a plane surface and is surrounded by a gaseous medium under pressure, illustrates the fact that, unless very special means have been employed to get rid of it, there will be a film of the gas between the two solid bodies; and this film is here shown as exerting an upward pressure equal to that upon the top of the plate, so that only the weight of the latter comes upon the supporting plane. Case B, on the other hand, shows the plate covering a hole of almost its own size, with only a very narrow

contact-surface around the opening; then the full unbalanced pressure upon the plate will come upon this narrow contact-strip, squeezing out the gas completely and altogether preventing leakage. Case C, representing the slide-valve, lies between A and B. With wider contact, the surfaces are less strongly squeezed together by the downward force upon the plate; the gas (or steam) is therefore given a chance to insinuate itself between the bodies and a film once formed will not only balance a part of the pressure upon the top of the plate, but will tend to flow. With this flow and leakage will go a drop in pressure from the high- toward the low-pressure edge of the contact-surface. Positive and exact information as to the amount and the manner of variation of this intermediate pressure is wholly lacking. We can only realize that there is a buoyant tendency, so that the downward force is less than the product of the unit pressure-difference by the area of the plate, or this whole area is not effective; and that the plate or valve must be pretty strongly held down if leakage is to be prevented. Obviously, the perfection of the surfaces, as to identity of form and as to smoothness, and their relative movement, are important elements in determining this action.

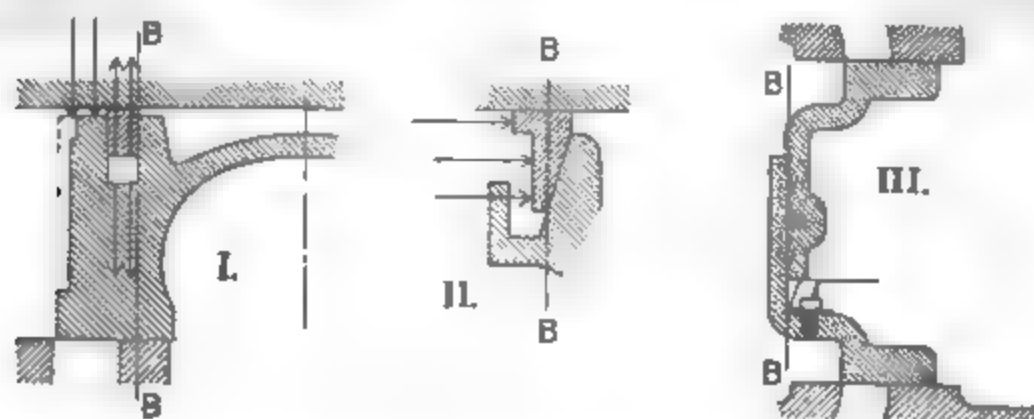


FIG. 421.—Boundary of the Relieved Surface.

*Discussion of Actual Valves.*—Turning now to the actual valve, we see at once that the area relieved of pressure must be very considerably less than that of the valve-face, to allow for a buoyant action of the steam under the valve and to insure plenty of force to hold the valve down. To get an idea of the proportions which experience has justified, we shall now analyze some of the examples that have already been illustrated. First of all, we must de-



line just what is the relieved area with any particular device. In Fig. 421 I., from Fig. 247, with the steam under the balancing-bars, the line BB very evidently marks the limit of the low-pressure area on top of the valve, which area is therefore the angle inside of the bars. Sketch II. is a detail from Fig. 409; with the ring held tight against the cone, direct downward pressure is relieved from the circle inside of BB—although the tendency of the ring to act as a wedge (very blunt, however) will produce some additional pressure, or move the effective position BB a little to the right. Detail III. is from Fig. 251, where steam is inside, and the location of BB is self-evident.

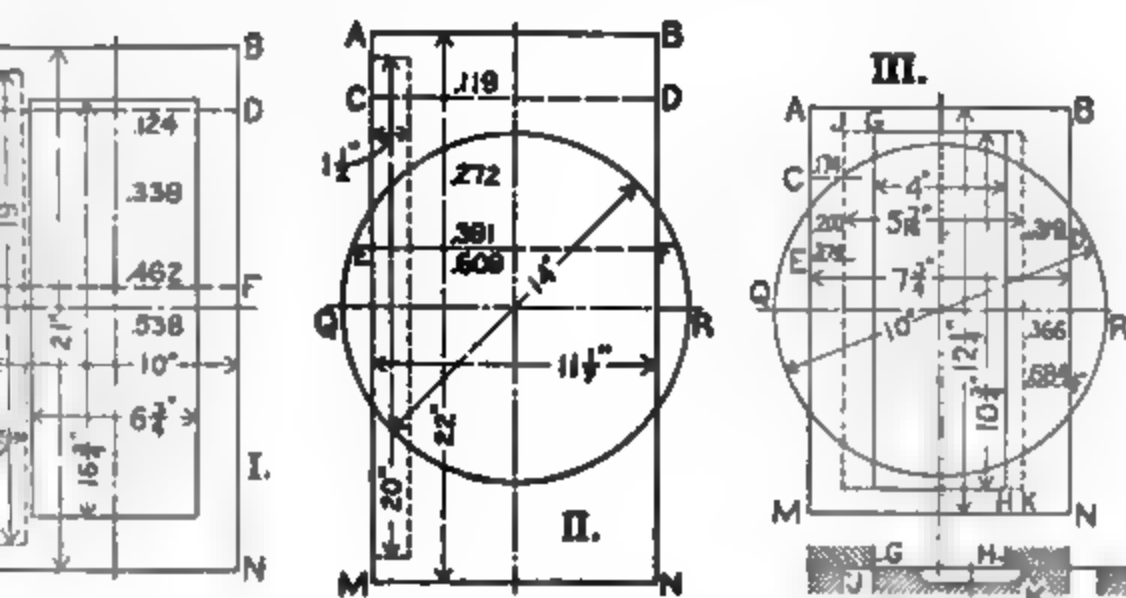


FIG. 422.—The Proportion of Relief.

Outline plan views of these three valves are given in Fig. 422. I., the total area of the valve-face MABN is 210 sq. ins.; the relieved area, shown by the inner full-line rectangle, is 113.1 sq. ins.; that is, of the whole area the fraction 0.538, represented by CFN, is relieved, while the rest is exposed to the full steam-pressure. But when one steam-port is open, as much of the valve overhangs its inner edge receives the upward pressure in the port—see Fig. 421 I. The limit of this action is the balancing of the whole area of one port, which is drawn in dotted outline at the left of Fig. 422 I., and diminishes the unbalanced area by 26.1 sq. ins. or 0.124 of the whole face, represented by CABD. Now the unbalanced area varies between 0.462 and 0.338 of the valve-

face, or between 96.9 and 70.8 sq. ins. Adding to this the area of the relief-ring (the bars are  $\frac{5}{8}$ " wide), which is 30.9 sq. ins., we have from 101.7 to 127.8 sq. ins. exposed to the steam-pressure. If the latter is only 150 lbs., and the coefficient of friction is 0.2 the resistance of the valve will range from 3050 to 3830 lbs. The action shown in Fig. 420 C will prevent this full maximum value from being reached; and with good lubrication the coefficient should be less than 0.2, though still very high as compared with the value usual in shaft-bearings.

Fig. 409, analyzed in the same manner at II., shows an unbalanced area ranging from 0.391 to 0.272 of the valve-face—the circle being taken to BB as located in Fig. 421 II. The valve from Fig. 251, at III., is more complicated. Without the rectangular openings in top and bottom, the steam would act upon the 78.54 sq. ins. of the 10-in. circle to force the halves apart. Subtracting the 43 sq. ins. area of the opening, we have 35.54 sq. ins. as the maximum unrelieved area. This is diminished as soon as the edge H passes K (see sectional view beneath main figure) and when the port is open the whole rectangle JK is relieved instead of only GH. The area of JK is 58.52 sq. ins., and the area subject to unbalanced pressure is now  $78.54 - 58.52 = 20.02$  sq. ins. The proportions of the balance are represented graphically, just as in the preceding cases: at the left E and C show, by their distance from A along AM, the ratio of the unrelieved area to the whole valve-face MABN; at the right, F and D show proportions in terms of the net valve-face, outside of the steam-area, or of the figure between AN and GH. This latter is the surface that has to be kept tight, and the effective holding area ranges from 0.684 to 0.366 of it.

*Marine-engine Relief-rings.*—The usual form of relief-ring for large marine-engine valves is typified in Fig. 423. In every case the "ring" is circular, is held in a slot in the valve-chest cover and slides upon the back of the valve. In the first example, the pressure which makes the ring steam-tight is secured entirely by the springs; in the other two the ring has a lip upon which the steam presses, to help hold it tight. The springs are made adjustable in the best practice, either by changing the little collar



in I., or by a set-screw as in II. Almost without exception, the sort of packing is used, to keep the steam from getting into the recess back of the ring, the wedge-rings in I. and III. being more effective than the plain snap-rings in II.

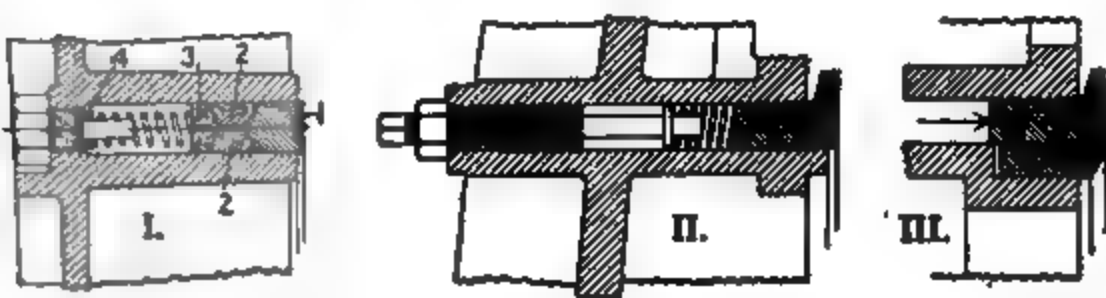


FIG. 423.—Detail of Relief-rings for Marine-engine Valves.

(g) **THE TWO-FACED FLAT VALVE**, rigidly balanced—see Figs. 6 and 7, 250, and 414—has the great advantage, as to ease of movement, that its tightness is secured and maintained by a close mechanical fit and not by unbalanced pressure. In general, it is a more delicate contrivance than the type with flexible balance, requiring a very high grade of workmanship in its fitting and adjustment: but when properly made and not subjected to too severe conditions of service, it is highly effective and will run for a long time without appreciable wear. Since the total clearance between the valve and its confining surfaces should not exceed a few thousandths of an inch, spring of the balance-plate must be reduced to a minimum by making it very stiff; and can be allowed for by very slightly arching the face when free. Irregular expansion must be avoided by warming up the engine gradually when starting. Practically, a long valve like that in Figs. 6 and 7 is of no use to relieve excessive pressure in the cylinder—even though it is not confined mechanically—because the area of the balance-plate is so many times that of the cylinder-port that a tremendous pressure in the latter would be required to lift the valve.

An interesting combination of the form of the two-faced valve with a flexible balance-rig on the principle of that in Fig. 409 is illustrated in Fig. 424. Here the pressure-plate 1 “floats”, being accurately located as against horizontal displacement by the two centering rings 3, 3, which are solid (uncut) and fit into corresponding recesses in the pressure-plate 1 and the balance-plate 2.

The top of plate 1 is relieved by the large ring 4, which is smaller in area than the whole valve-face, so as to insure enough downward pressure for tightness. To compensate for the extra area on the bottom of this plate which is exposed to high pressure when the port is open, a small relief-ring 5 is placed over each port, receiving steam on the inside, from the port, through several good-

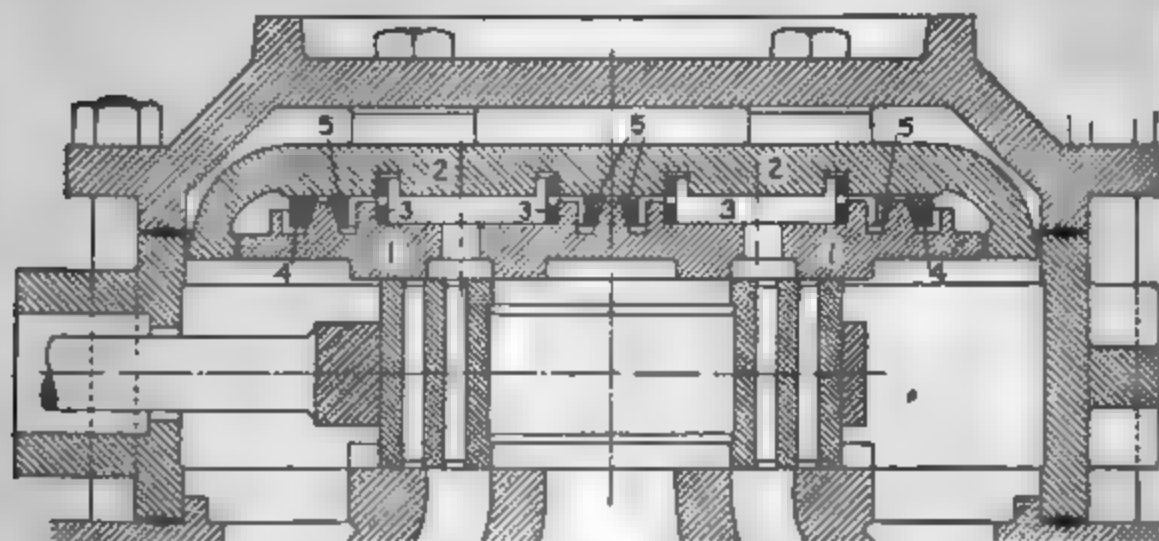


FIG. 424.—American Balance Slide-valve Company's Two-faced Flat Valve for Locomotives.

sized vent-holes. The inside of the main relief-ring is freely vented to the exhaust-port.

The advantages of this design are that double port-opening is secured with an arrangement which does not require precise fitting, which automatically takes up wear, and which lets the valve rise freely to relieve the cylinder—a property very useful when the locomotive is “drifting”, or running with the steam shut off further, a long valve can be used, if desired, with shorter steam-ports. But since as much unbalanced pressure must be allowed to insure tightness, as with an ordinary valve performing the same functions, and since there are two surfaces where friction will act, it appears that this valve is, on the whole, not any better than an Allen valve with an equivalent relief-rig on its back.

(h) PLAIN PISTON-VALVES. Examples of simple four-function valves, controlling one cylinder, have been given in Figs. 252, 254 (high-pressure cylinder), and 261. The valves first chosen here for detailed illustration, Figs. 425 and 426, are from the loco-

re. In both cases there is a central body and two heads, held together by the valve-rod alone, and a bull-ring carrying two

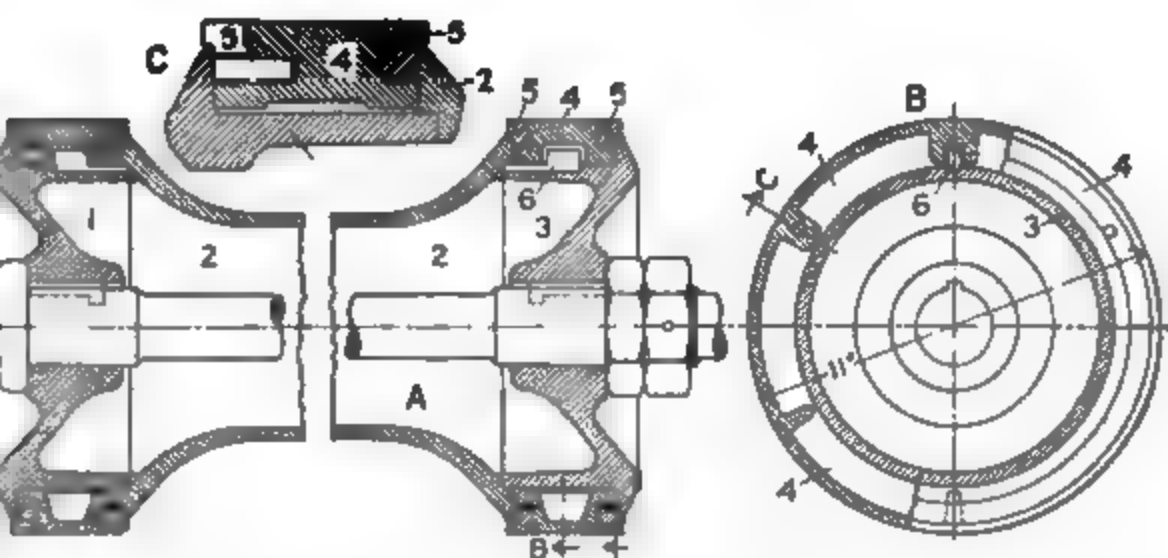


FIG. 425.—Valve with Closed Ends, for  $20\frac{1}{2}'' \times 28''$  Express Locomotive.  
Scales 1 to 8, 1 to 4.

map-rings with projecting edges, to form the working-face at each end. Note how the bull-ring is kept from turning, by a key in Fig. 425, by a riveted pin in Fig. 426, as shown at B; and how pin-keys are likewise used to keep the ring-joints at a definite position, two ways of cutting the rings being shown at E and F in Fig. 426. With the further keying of the valve-heads to the

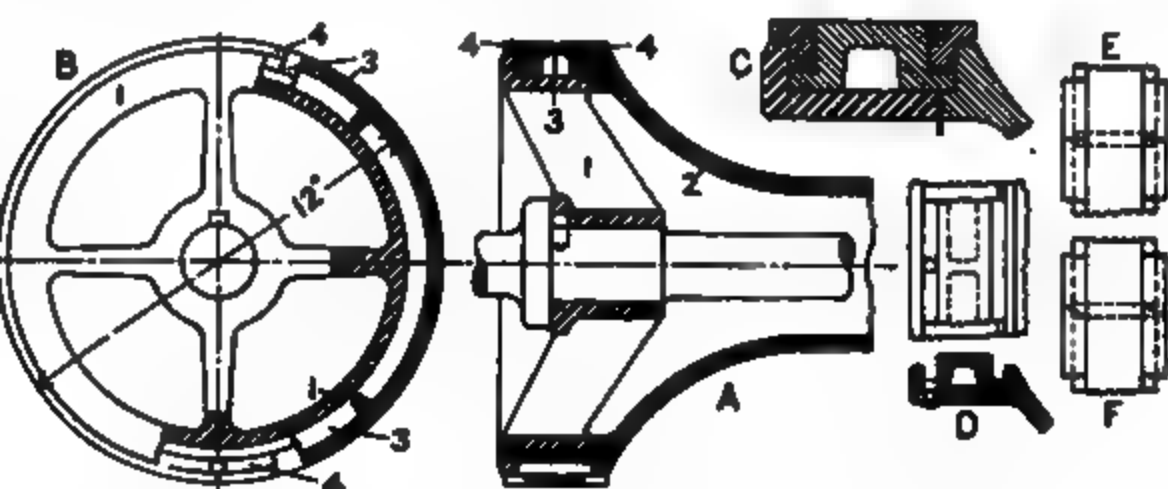


FIG. 426.—Valve with Open Ends for  $20\frac{1}{2}'' \times 28''$  Express Locomotive.  
Scales 1 to 8, 1 to 4.

rod, the location of the ring-joints is positively determined. In Fig. 425 the bull-ring is just fitted neatly to its slot, with some

freedom for self-adjustment, but in Fig. 426 it is tightly clamped, fitting closely upon the head and serving to hold the latter in line with the valve-body. Rings resembling those in Fig. 280 XIII are used in the first valve; but the shoulder has so much clearance that it is evidently intended to confine the ring in case of breakage, rather than as a restraint against undue expansion in ordinary running. Both valves are of the inside-admission type; and there must be an exhaust-passage at each end, the two meeting below the exhaust-nozzle, within the saddle-casting. The passage through the valve in Fig. 426 helps to equalize the exhaust-pressure in the two ends of the valve-chamber, but is not large enough to carry the whole current of steam as in some arrangements.

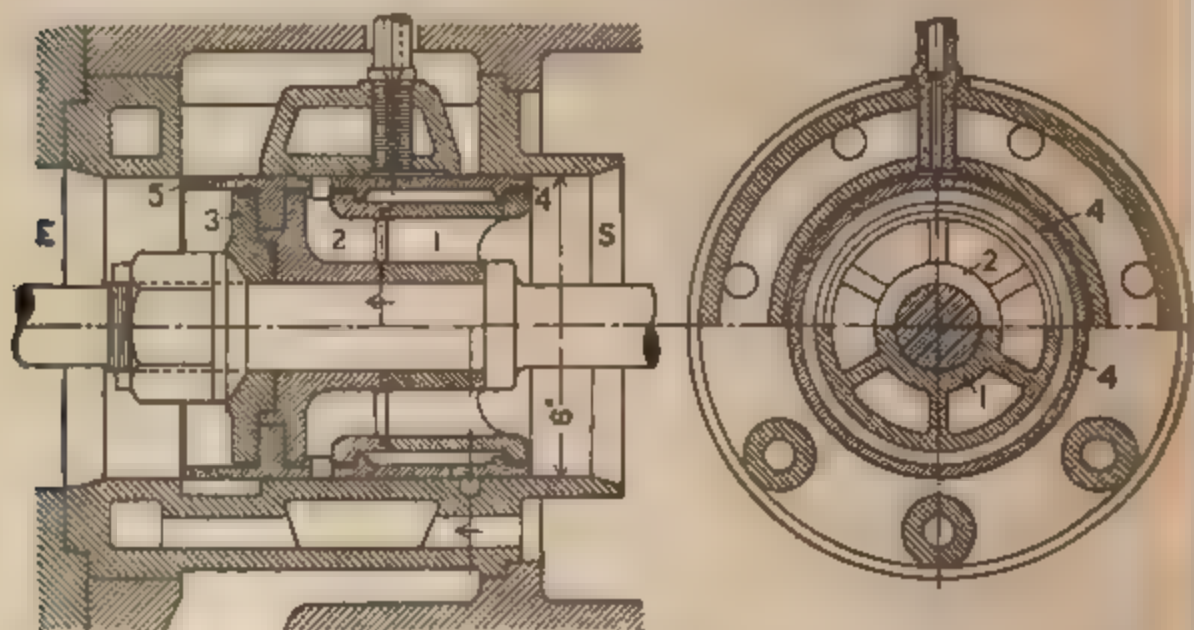


FIG. 427.—Solid-faced Valve, for German Locomotive using Superheated Steam.  
Scale 1 to 6.

*Small Solid-faced Valves.*—The use of ordinary cast-iron packing-rings to make piston-valves tight is very common—as witness Figs. 255, 258, 435, etc. Small valves, up to six inches in diameter or a little more, are often made solid, resembling the two-faced flat valve in being just loose enough to move easily, without permitting leakage. An entirely simple valve of this type is shown in Fig. 252, another of more complicated form in Fig. 434. One-half of a solid-plug valve used with superheated steam is drawn

detail in Fig. 427, where the working-face is made up of two  
ht shells, held in a head formed of three pieces recessed together.

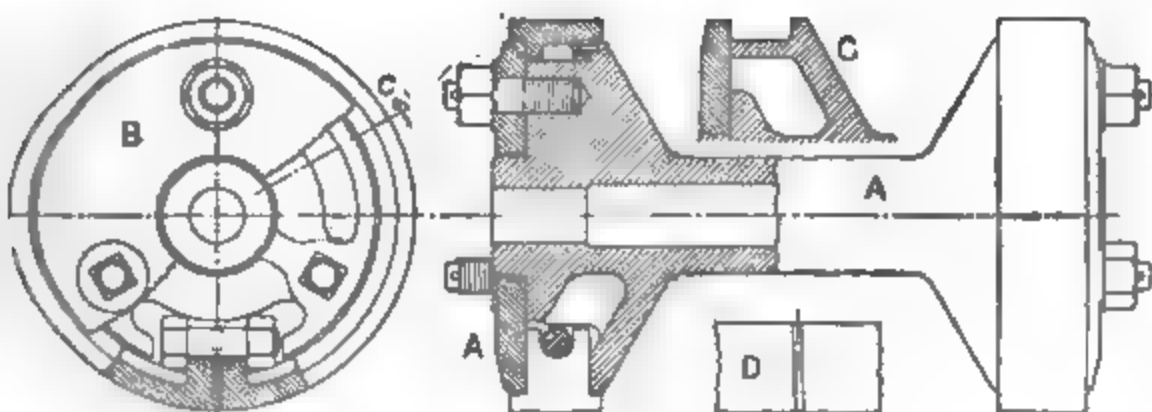


FIG. 428.—Valve with Adjustable Face-ring, Detail from Fig. 260. Scale 1 to 6.

he shells are grooved to hinder leakage, and the oil for lubrica-  
on is applied directly to the valve, through the pipe at the top.

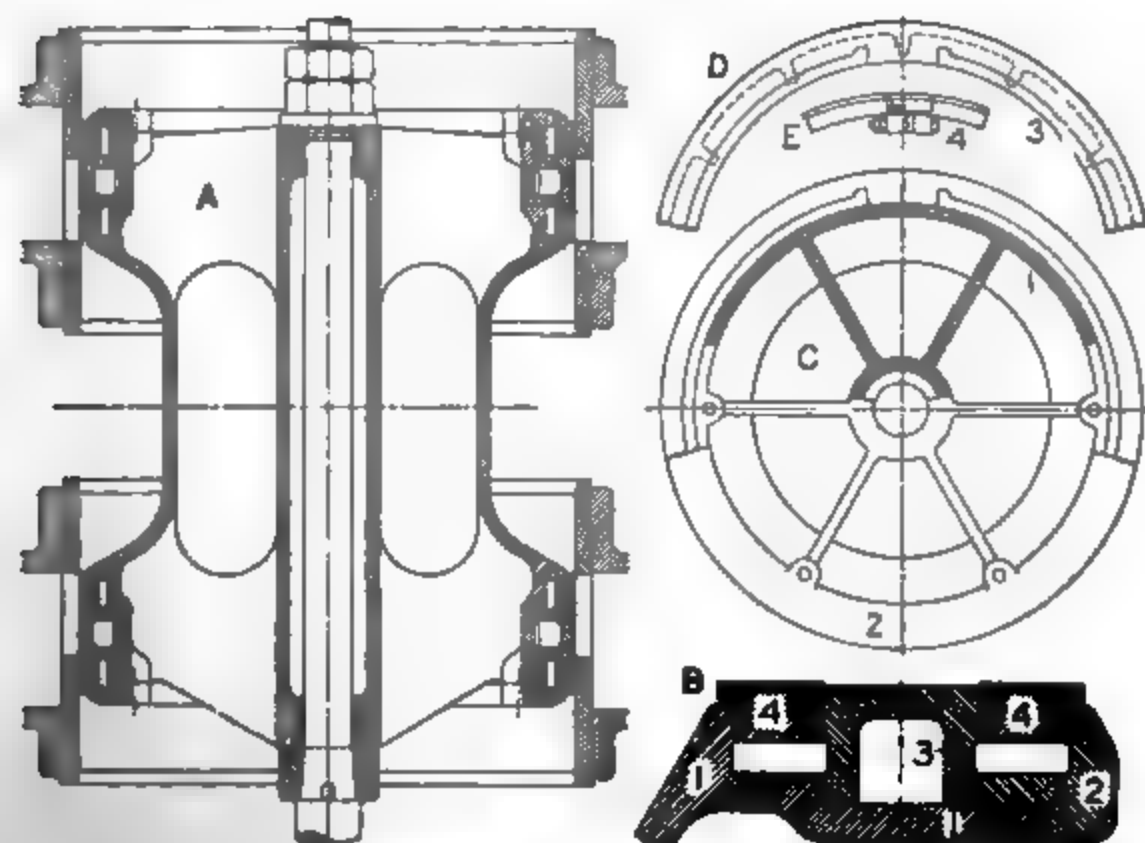


FIG. 429.—Large Piston-valve, of the Best Marine-engine Type.

To insure uniform heating, the valve-casing is made hollow and is  
open to free circulation of the live steam. As regards its action,

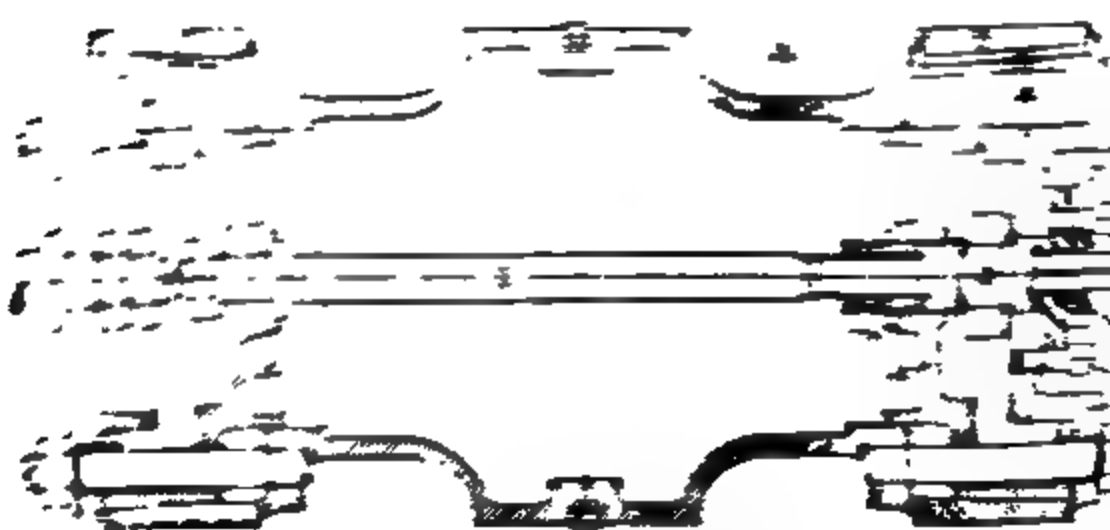


Fig. 254 — Valve with Tapered Stem. Detail from Fig. 254.  
 (Enlarged Section of Valve Head.)

; and it is to be noted that the valve can be added as the valve wears low.  
 In Fig. 254 is shown a large marine-engine valve, with an arrangement of working-face best shown by the enlarged section at

detail of the ring-joint is seen at E, and the way that space provided for this joint at C and D. In view C the bull-ring 3 removed, and we see the recess for the joint of the inner ring; the D shows how it is made for the outer ring. This large hollow valve furnishes ample passage through its middle part, so that a stem-connection need be made at only one end of the valve-member.

Fig. 430 shows a small solid valve, for a horizontal high-speed compound engine arranged like the locomotive in Fig. 257—only the half (one end) of the valve being given. It is designed on the idea that the wear of the valve and seat is due wholly or chiefly to the weight of the valve; so that dividing the valve lengthwise in a horizontal plane, and inserting thin copper liners as needed, will enable a good fit to be maintained.

*Expandable Valves.*—Another type of device is well represented in Fig. 431, where the rings 5, 5 are set out by cone-wedges to any desired degree of tightness. The top of the valve, as it stands in the vertical engine, is here at the right; and both working-faces can be adjusted from this end, without removing the valve from its casing. The hollow threaded spindle 7 can turn

in the head 3 without lengthwise movement, and can be locked by the cap-screw 8; while the long spindle 6, threaded into 7, can be locked with 7 by the screw 10. The lower cone 4 is moved by turning 6 in 7, the upper cone 4 by turning 6 and 7 together. The ring-joints both come at the upper edge of view A.

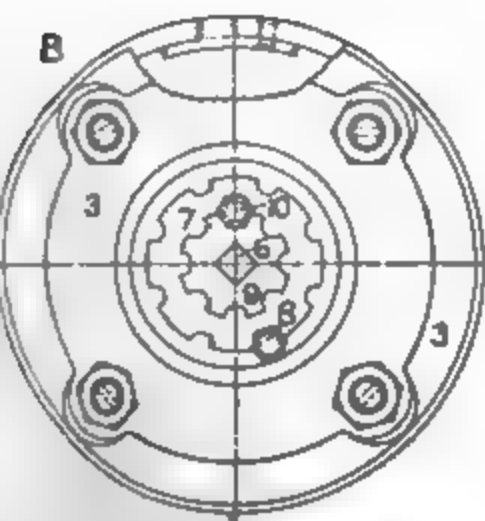


FIG. 431—Continued.

is adjusted instead of the valve. This bushing is slipped into place from the exhaust side (the left in view B), and the ring which holds it is fastened with cap-screws. The valve has three "fingers" which project inward from the steam edge, fitting closely

to the valve-seat; and one of these covers the gap at the joint of the bushing. When the screw-cap above the end of the clamping screw is removed, the latter can be turned with a small screw wrench, and the adjustment made with steam turned on if desired.

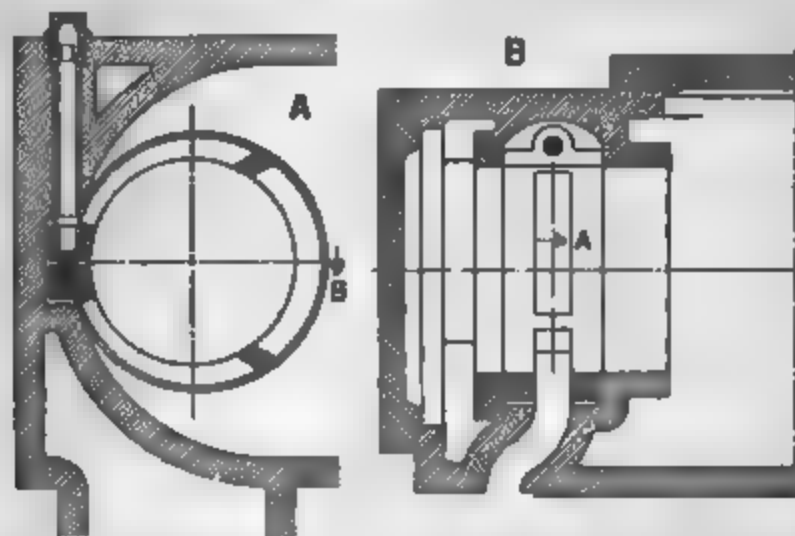


FIG. 432.—The McIntosh & Seymour Adjustable Valve-bushing.

*A Self-adjusting Valve.*—A device which automatically adjusts and clamps itself is shown in Fig. 433, the underlying principle

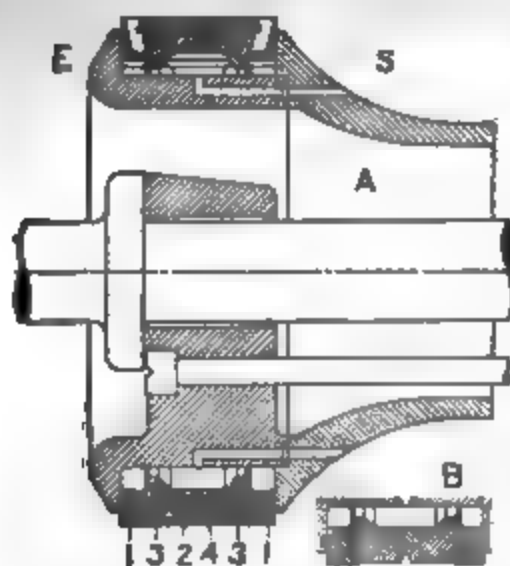


FIG. 433.—American Balance-Valve Company's Semi-plug Piston-valve.

being the same as in the piston packing arrangements at XV. XVI. in Fig. 280. There are first the two snap-rings 1, 1, and the lap-plates at their joints as shown at the top edge of this section view. With these is combined an expansible face-ring 2, which can, however, easily be omitted, and change to the form at B. The rings 3, 3 are solid; and between them is a wedge-ring 4, which like the packing-rings has a plate at the joint. Through a number of drilled holes, live steam

is admitted back of the whole system of rings: when first turned on, it pushes out the rings 1, 1, till they make good solid contact with the casing; then ring 4 wedges 3 and 3 apart, holding 1



against any further tendency to expansion. The light screw shows the valve-rod is to hold the parts together when the rod is removed.

(j) **MULTIPLE-PORTED PISTON-VALVES.**—Sometimes the piston-valve is made like the Allen valve in Fig. 410. An example is given in Fig. 434, where the valve has inside admission. This also shows a method of construction sometimes used for small valves, the two cast-iron "pistons" being joined by a piece of common wrought-iron pipe, screwed in—which greatly simplifies the work of the foundry in casting the valve.

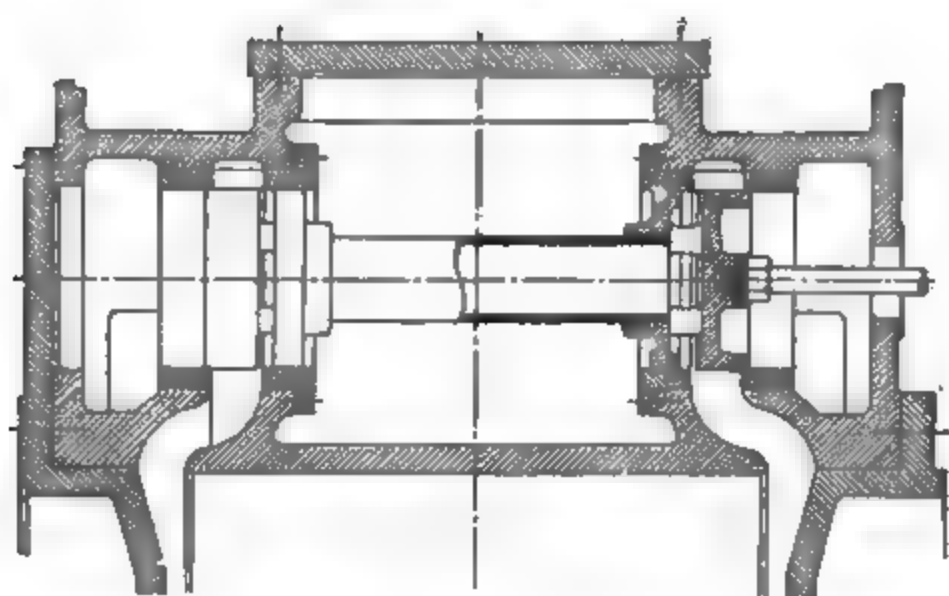


FIG. 434.—Piston-valve of the Allen Type.

Another double-admission valve has already been shown in Fig. 427: and a large valve, for the low-pressure cylinder of a two-cylinder compound locomotive is illustrated in Fig 435. The vertical section A shows clearly the action of the valve, while the horizontal section in two planes at B and the cross-sections at C bring out the form of the casting and show how the extra steam-passages open toward the ends of the valve and the extra exhaust-passages toward the middle, without interference.

(k) **COMPOUND VALVES.**—This name seems the most suitable for valves which control the steam-distribution in two cylinders—such valves being possible in engines where the two pistons move together or are connected to cranks at 180 degrees. Examples have been given in Figs. 254, 255, and 258: and another, for a

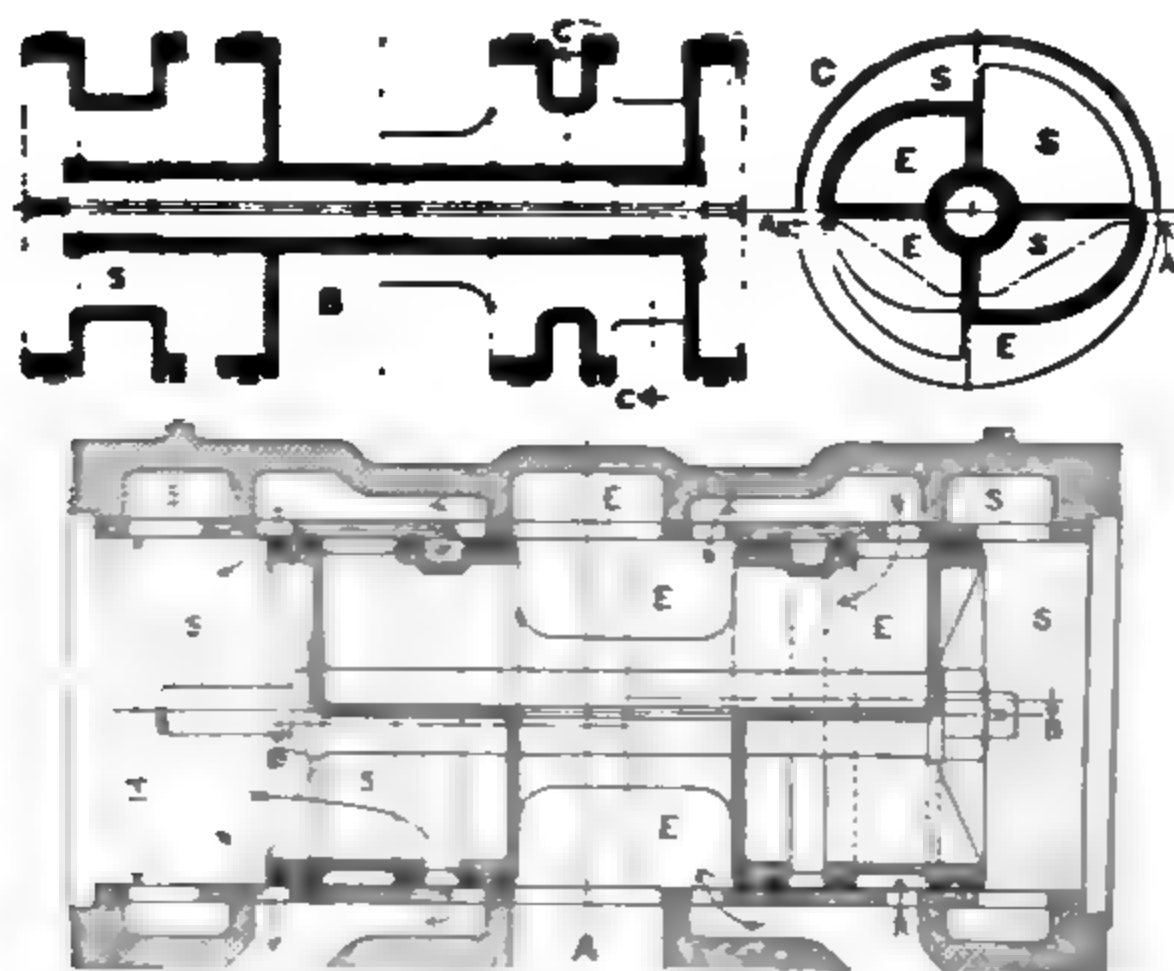


FIG. 432.—24-ported Valve for 35" x 26" Low-pressure Cylinder of a Compound Locomotive. Scale 1 to 12.

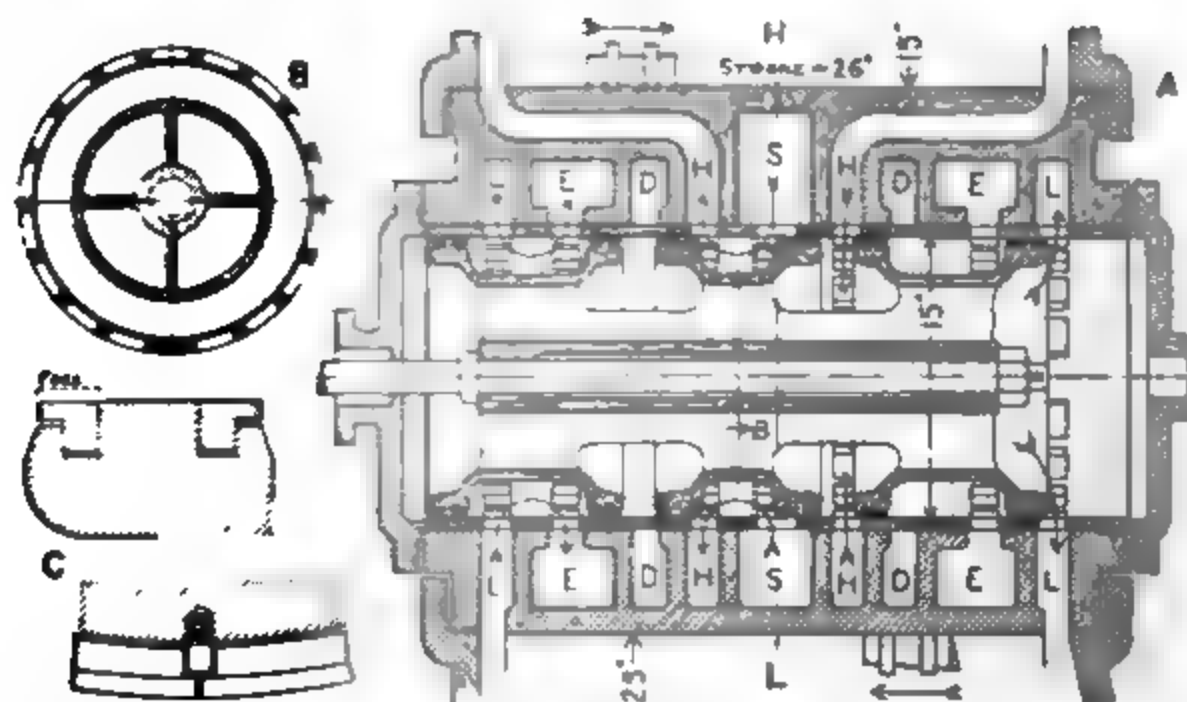


FIG. 433.—Valve of Vaucain Balanced Compound Locomotive. Scale 1 to 16 and 1 to 3.

locomotive much like Fig. 256 in cylinder-arrangement, but with cranks opposite, is fully shown in Figs. 436 and 437.

This locomotive has two high-pressure cylinders inside the frames and two low-pressures outside, with the two cranks at each side set opposite, and these pairs quartered, as in Case B, Fig. 40 (k). The valve is shown at about its extreme distance toward the left. Its middle "spool" controls the H.P. cylinder, with steam inside and exhaust outside. The central cavity of the

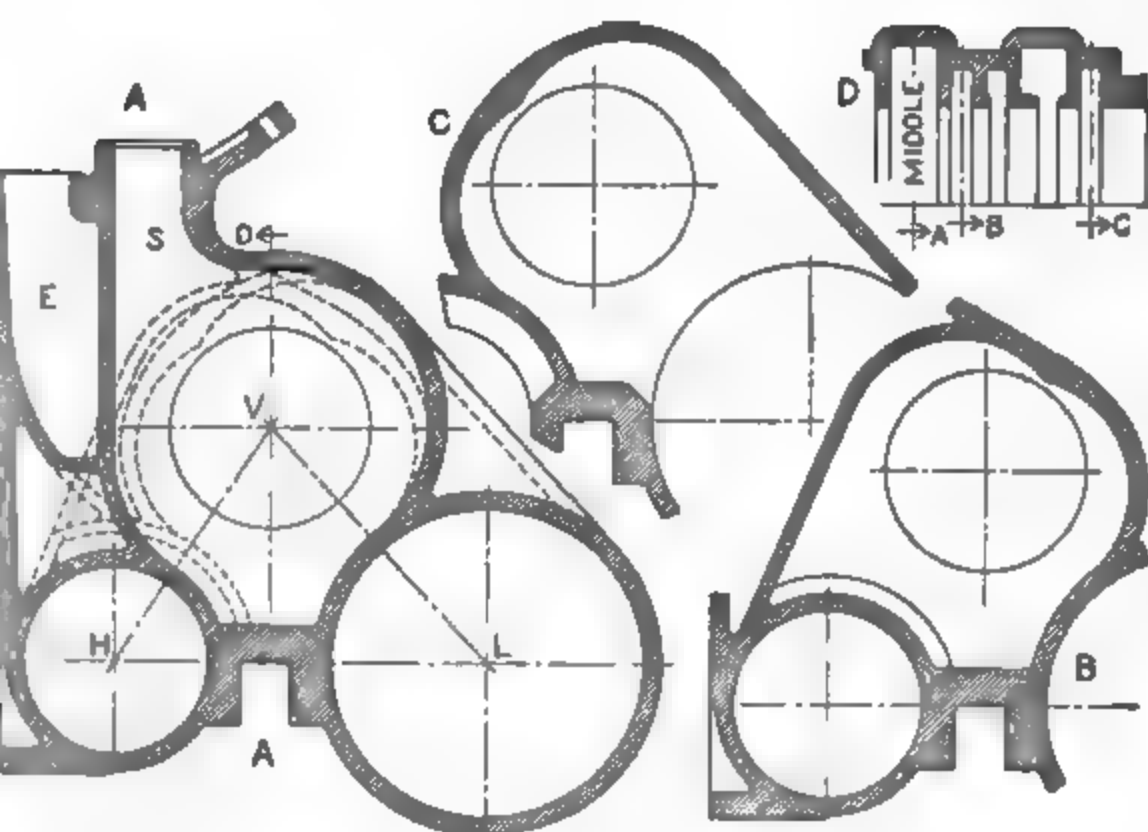


Fig. 437.—Sections of Cylinder and Valve-chamber of Vaucain Locomotive—with Fig. 436. Scale 1 to 16.

valve—that is, the whole valve-chamber—acts as a receiver between the cylinders, the working-faces at the ends of the valve controlling the L.P. cylinder. Then the faces at the inner ends of the outer spools do not control any ports, but simply serve to enclose the pockets through which exhaust takes place; and the chambers marked D, outside the valve-bushing, are mere dead spaces, hollowed out in the casting. Details of the rings are shown at C, and the superiority of rings of this type over the plain rectangular rings in Fig. 258 is apparent, in that they furnish better working-

edges and are more definitely pressed outward by the steam behind the projecting lips.

Sections of the cylinder-casting are given in Fig. 437, views A, B, and C being made by cutting-planes located as shown at D; that is, A is a middle section, B follows the high-pressure port, and C the low-pressure port. In Fig. 437 half-sections along VH and VL are swung into one plane. It is evident that the clearance-volumes are quite large.

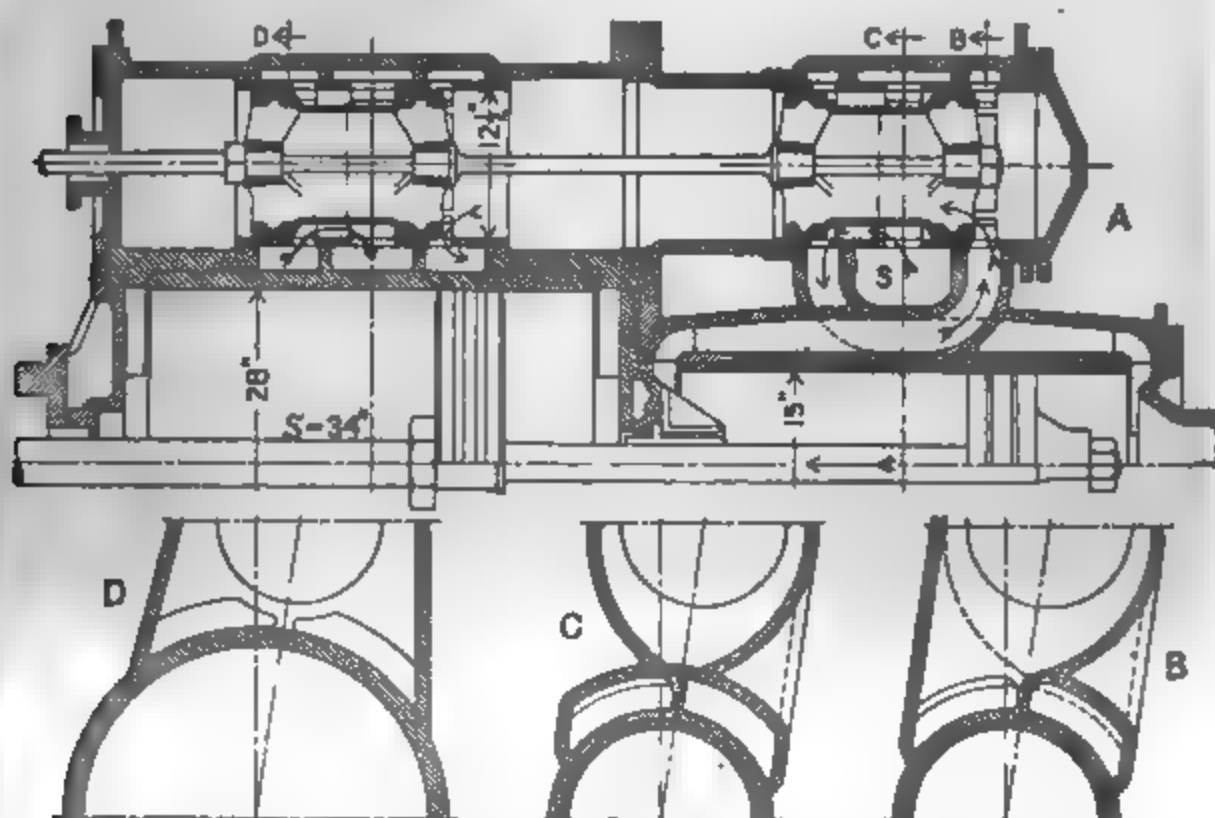


FIG 438. Valves of a Tandem-compound Locomotive. Scale 1 to 24.

An interesting arrangement in which two complete valves are combined on one valve-stem, so as to control two cylinders with the same movement, is shown in Fig. 438. Since steam must be admitted to both front ends or both back ends at the same time, it would seem proper, at first sight, to use two valves of the same type, either both direct or indirect. Actually, with a view to simplicity in the steam-passage and receiver between the cylinders, it is better to use valves of opposite types and to cross the high-pressure ports. There is plenty of room for these ports; how they pass each other is clearly shown by the

tions at B and C; and the only drawback to this arrangement is the increased clearance in the high-pressure cylinder. It is desirable that the valve-axis be as near the cylinder-axis as possible, wherefore the low-pressure port is crowded against the cylinder in the middle, but spreads out to its full thickness on each side of the valve-chamber, as shown by section D.

In this connection it is most appropriate to illustrate the Willans central-valve engine, which is shown in its compound form in Fig. 439. The piston-rod is a tube of considerable size, within which slides the multiple-piston valve. This valve is driven by a fixed eccentric, forged upon the crank-pin, so as to receive the proper relative motion. Of the numerous pistons on the valve-rod, only the two marked  $V_1$  and  $V_2$  really act as valves to control the flow of steam through ports in the casing; the others are merely movable partitions, dividing this valve-chamber so as to form the needed passages from port to port. Further, the valve proper controls not only admission and the two exhaust-vents; cut-off is effected by the closing of the ports marked  $P_1$  and  $P_2$ , as they slide into the packing-boxes in the cylinder-heads — and with the full speed of the piston to close these ports, cut-off is very sharp. For symmetry of force-action, the connecting-rod is in duplicate, a rod being placed on each side of the eccentric-rod. The guide-piston  $G_2$  serves as cross-

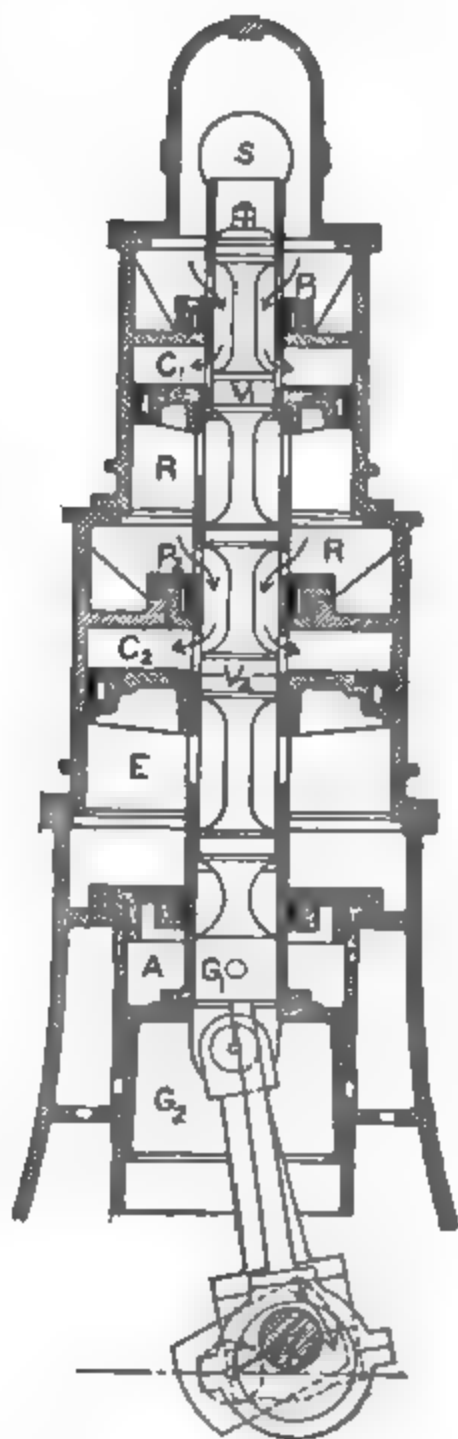


FIG. 439.—Willans Central Valve.

head of the eccentric-rod. The guide-piston  $G_2$  serves as cross-

head; and to insure that there shall always be a downward force upon the connecting-rod, air is compressed in the chamber A. At the very high speeds usual with this engine—400 to 500 R.P.M. in the smaller sizes—the large moving mass will develop very great inertia forces; and the clearances are so small that but little steam-cushion is permissible. Two or three units like Fig. 439 are placed closely side by side, acting on one shaft; and frequently each unit is a triple-expansion engine. The peculiar action of the receiver in this engine is discussed in § 69 (m).

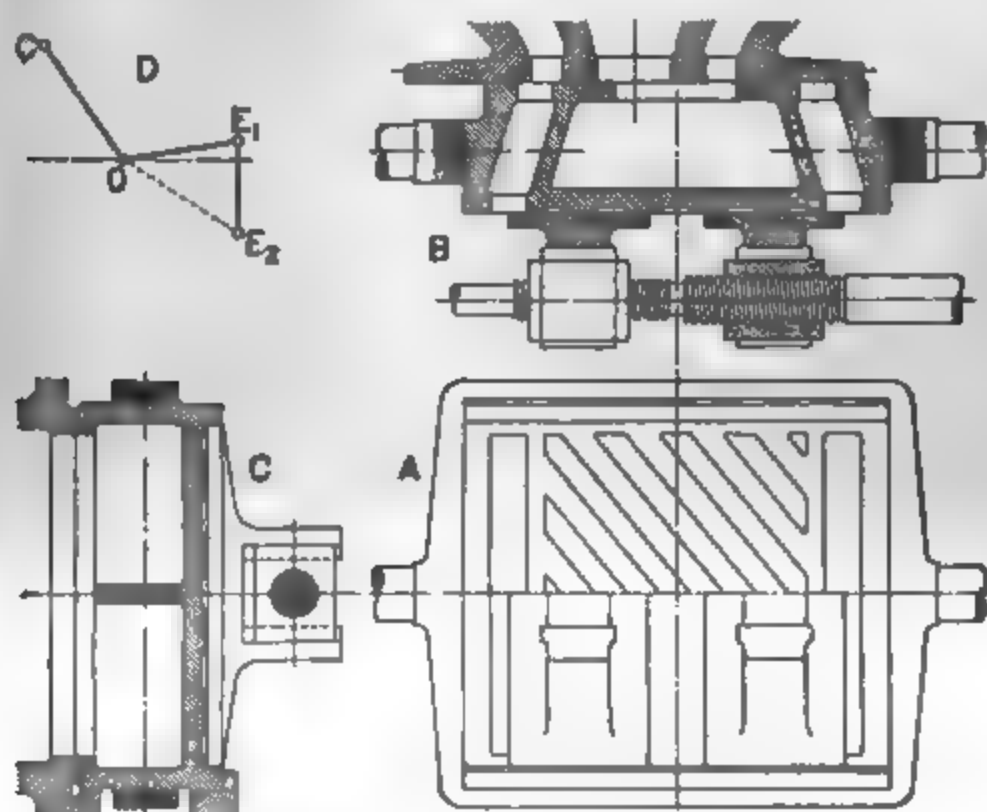


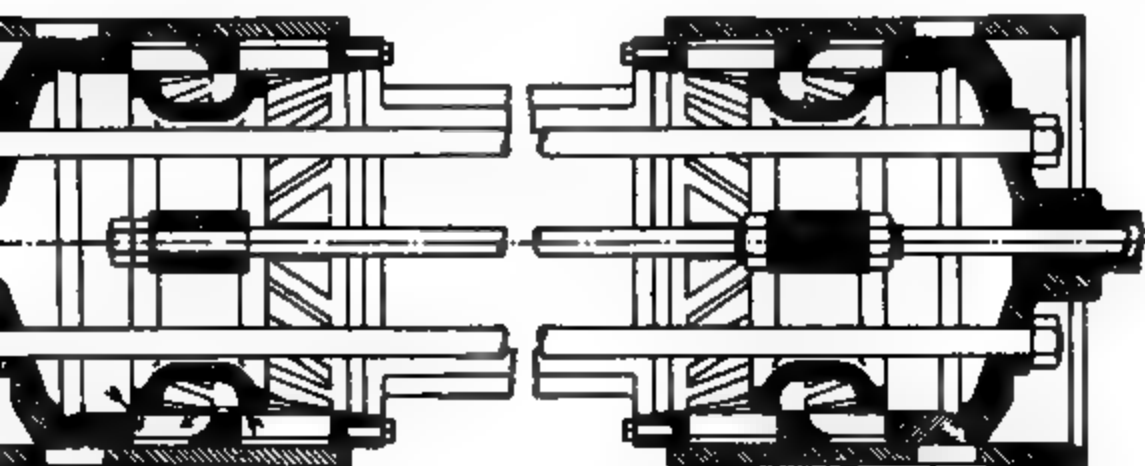
FIG. 440.—Detail of a Meyer Valve.

(l) DOUBLE VALVES — A good example of the Meyer valve is given in Fig. 440, shown with more detail than the one in Fig. 398. A novel feature of this design is the corrugation of the back of the main valve in gridiron fashion, which gives the steam free access to the bottom of the cut-off slides, and thus prevents these from being subject to an unbalanced pressure during their idle movement. The valves are drawn in positions corresponding to the crank-eccentric diagram at D.

Fig. 441 shows a double valve of the piston form, similar in

ral idea to that in Fig. 253, but with the inner valve formed to give double admission—this valve being controlled by a governor. The design has a number of interesting details, the construction of the main valve, and in the device for adjusting the laps of the cut-off valve by moving one of the pistons on rod.

o far we have seen two types of the double-valve gear, the er, in which the laps are changed by moving two short slides t or together, and the other type, without a convenient dis- ive name but exemplified in Figs. 404, 405, and 441, where rigid second valve is moved by a changeable eccentric. Another e, which secures variable lap with a solid valve, is the Rider,



441.—Double Piston-valves, for 44" × 48" Cylinder of Rolling-mill Engine—Wm. Tod Company. Scale 1 to 18.

l represented by Fig. 442. Here the ports in the back of the in valve are made oblique, which gives the casting quite a complicated form, but enables the lap to be changed by simply moving the cut-off valve sidewise. In view A, the right half the riding valve is removed, to show the main valve more clearly: it appears that the idea of providing a bearing-surface for the second valve only near the ports is here carried out much more ly than in Fig. 440. The arrangement for giving the expansion ve its sidewise displacement is clearly shown in views B and and it is obvious that this device, requiring only a small angular vement of the valve-rod, can be much more easily placed under e control of an automatic governor than can the Meyer gear.

The Rider valve is quite often given the piston form, the oblique ports then becoming helical; this complicates the main-

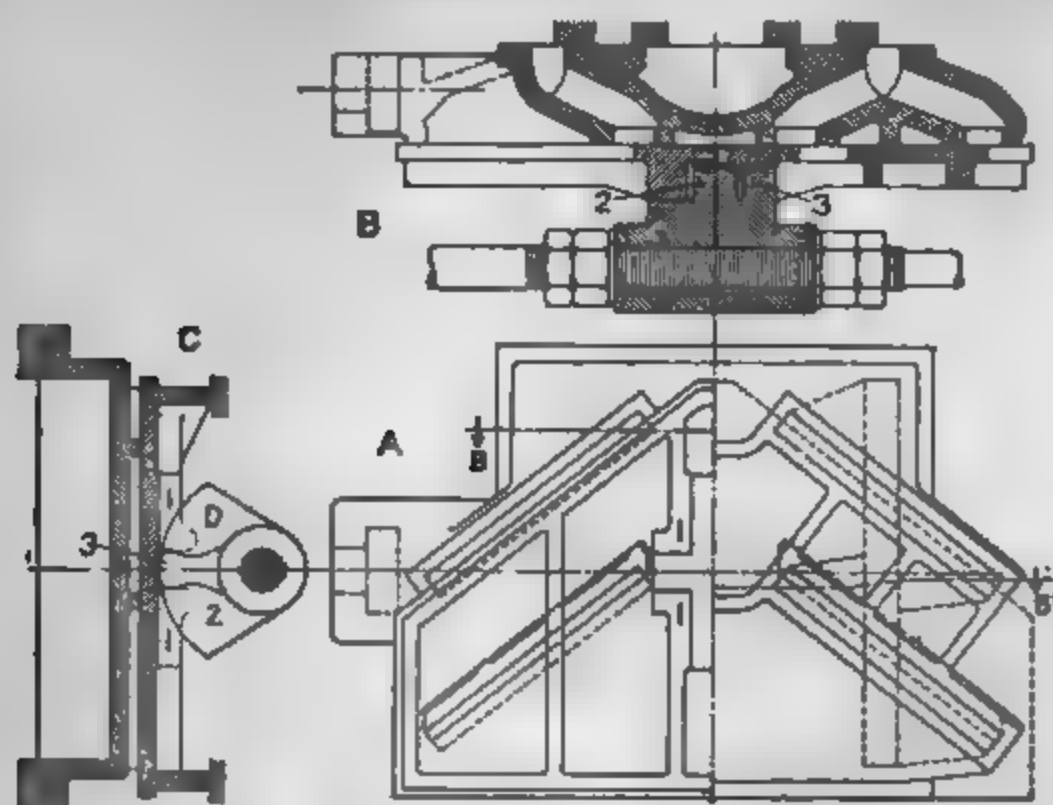


FIG. 442.—Rider Valve with Flat Seat—German Design. Scale about 1 to 10.

valve casting, but otherwise is a simple adaptation of Fig. 442, with a great gain in the balancing of the valves.

### § 55. Valve-gear Details.

(a) THE ECCENTRIC.—In its more usual form, this piece is the pin of a short crank, enlarged into a disk big enough to go around the shaft. We shall here consider fixed eccentrics, as distinguished from those carried by shaft-governors. A general view of practice, well represented by the examples given in Figs. 443 to 449, may be summed up in the following statements:

*Form of the Eccentric.*—The wider side of the disk or "sheave" is very much in the form of a wheel, with one or more arms, Fig. 368 showing an extreme case. The plainest form of rim and hub is seen in Figs. 443 and 444; in Figs. 445, 447, and 448 there is



central stiffening-web; while a full side-web is used in the double eccentrics, Figs. 446 and 449.

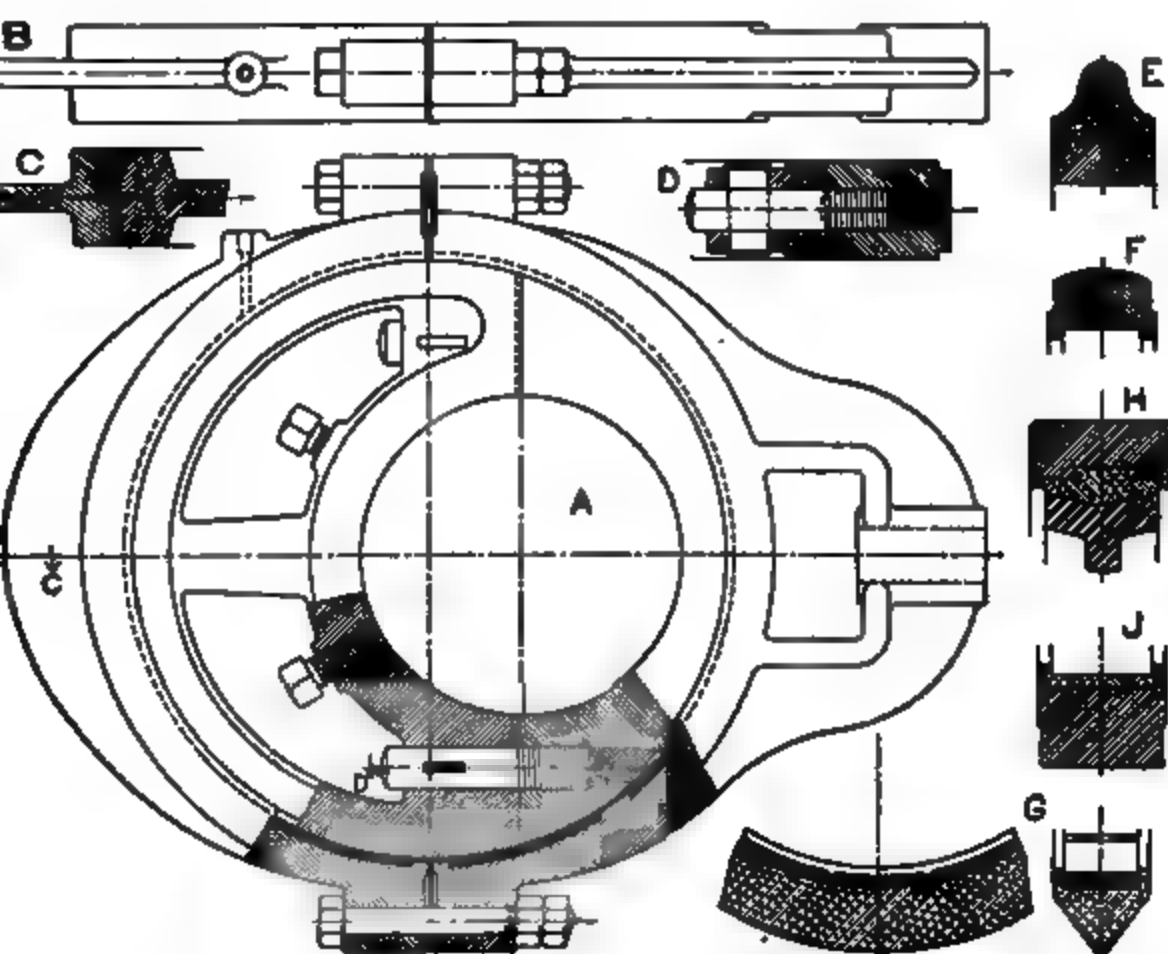


FIG. 443.—Eccentric for Corliss Engine, to fit shaft in Fig. 312 I. Scale 1 to 14.  
Sketches E to J, various sections of the eccentric-strap.

*Dividing the Eccentric.*—When the eccentric is to be placed upon the main engine-shaft, it is nearly always made in two pieces, held together by bolts with nuts or keys. Usually the joint is recessed, although sometimes fitted bolts are used, as in Fig. 445: the most positive joint has the tongue-and-groove running radially, as in Figs. 443 and 444. The clamping-bolts must be well secured against working loose; in Fig. 443, for instance, the key is driven in hard, then lightly riveted.

*Material.*—For stationary engines and locomotives the eccentric is made of cast iron, except that sometimes the smaller "half" is of wrought metal, for the sake of strength; but in the higher grade of marine engines, cast-steel eccentrics like Figs. 448 and 449 are very common. Occasionally, especially in very quick-

running marine engines, the eccentrics are forged solid with shaft—which reduces rubbing-speed and saves weight, but costly.

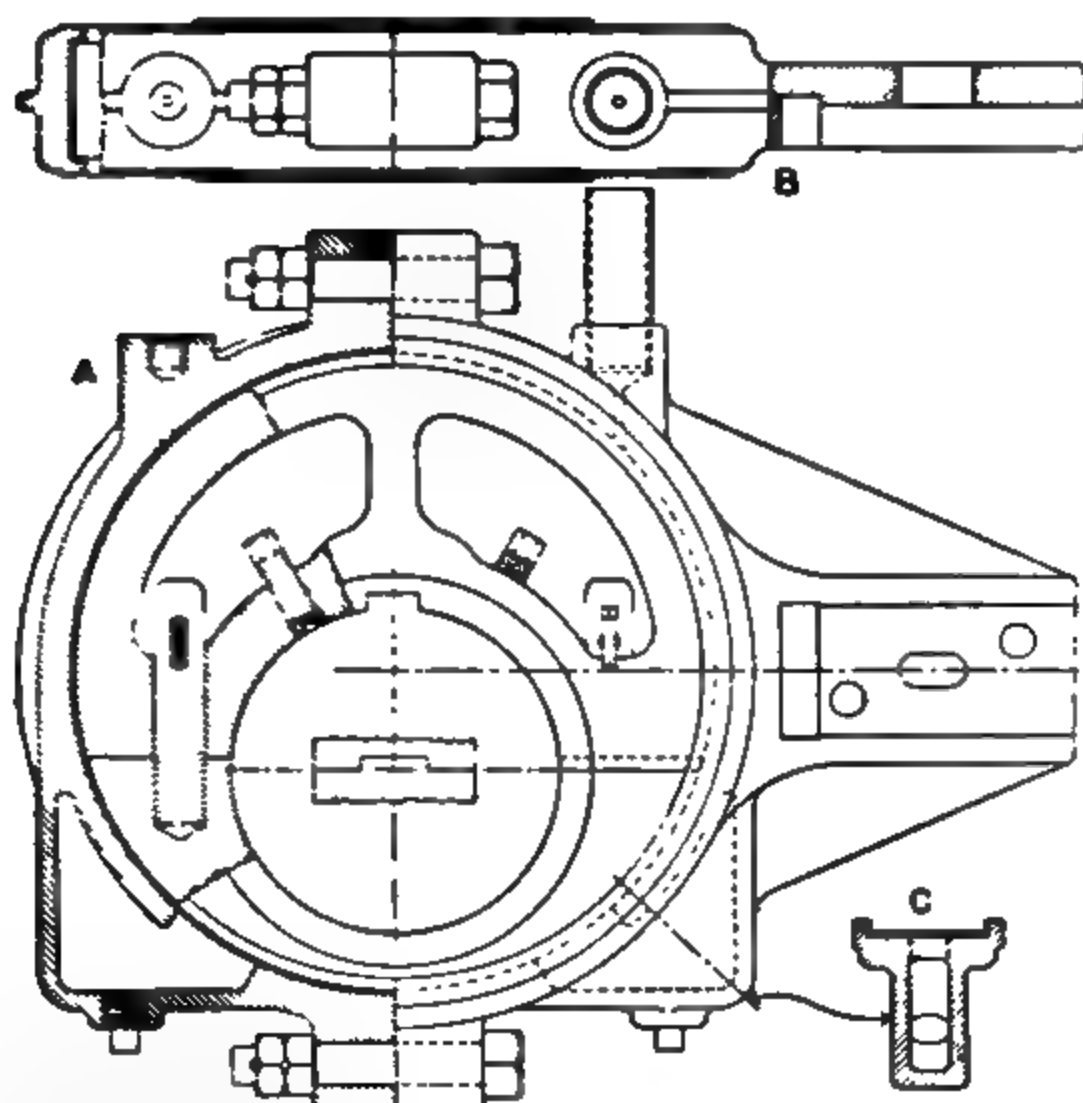


FIG. 444.—Locomotive Eccentric with Cast-steel Strap, Detail from Fig. 3. Scale 1 to 8.

*Fastening the Eccentric.*—In the link-motion the eccentrics are usually keyed upon the shaft, because the conditions of service are severe and a positive fastening is needed. With the early running Corliss gear, set-screws give sufficient security, accompanied by flexibility. Ease of adjustment is especially desirable where the exhaust-valves are separately driven and the engine is liable to such a change of condition as from open exhaust to running condensing. The eccentric-hub is bored to an easy close fit, so that it can be moved upon the shaft, but will have no motion. When set-screws are used along with a key, as in Fig. 444,

, they serve to take up all slack, besides preventing endwise movement. One reason for not using set-screws alone where the eccentric must be very strongly held is, that if they are screwed down very hard they are likely to distort the sheave.

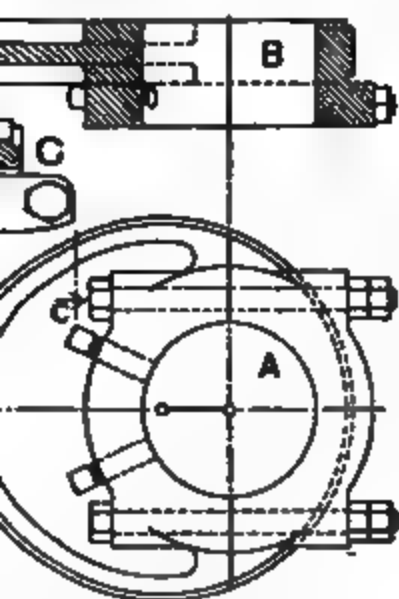


FIG. 445.—Eccentric with Side-hub, for Exhaust-valves of Simple Engine like Fig. 215.

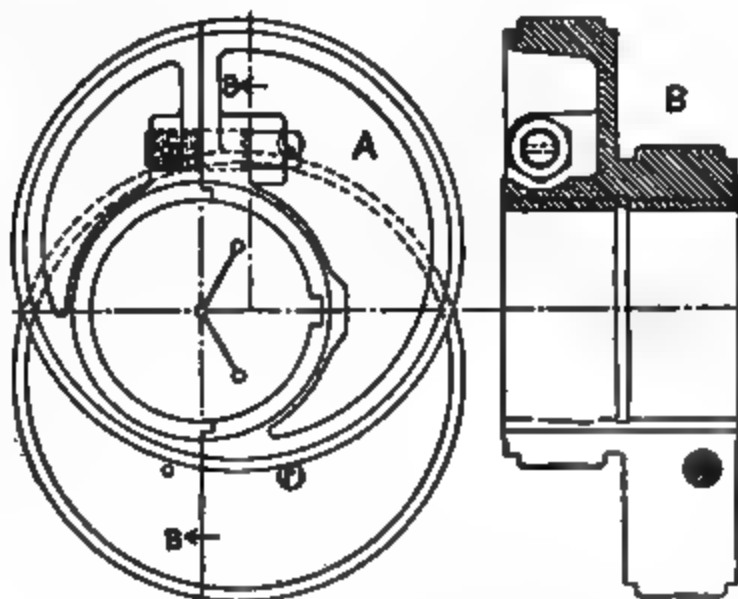


FIG. 446.—Double Eccentric, for Locomotive, see Valve-gear in Fig. 455.

*Adjustable Eccentrics.*—For large engines a simple set-screw tightening is rather crude. A better type of arrangement, frequently used, is represented by Fig. 447:

This is taken from a large vertical Corliss engine which has its valve-gear driven by a lay-shaft instead of the 26" main shaft—the two being connected by an intermediate vertical shaft with spiral gearing. The drawing shows clearly how the eccentric is clamped to the fixed

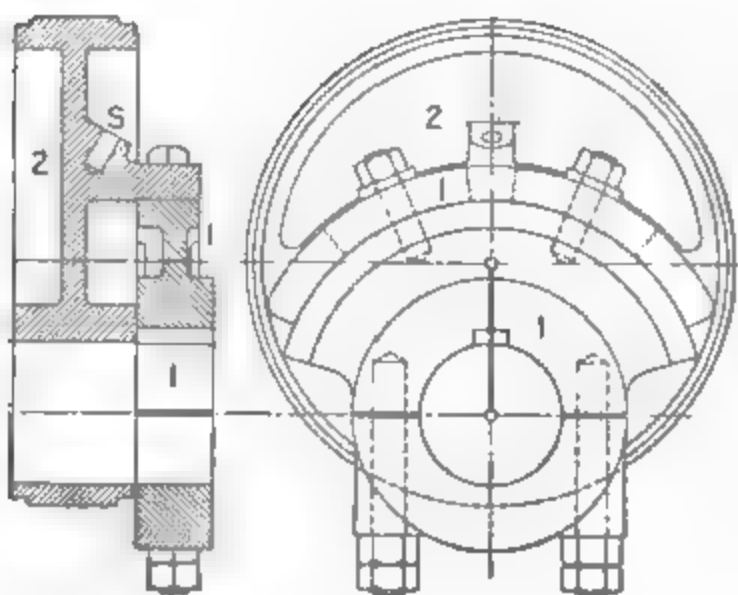


FIG. 447.—Adjustable Eccentric, Clamped upon Fixed Hub.

hub or clutch 1, by studs which pass through oblong slots. The

# VALVEGEARS AND THEIR ACTION.

eccentric is to provide a pin, by which the eccentric can be removed when the bolts are slackened off. Sometimes the eccentric is keyed fast and the other bolted to the crank pin in the same manner.

**Eccentricity.**—This is nearly always cylindrical, and the ends to make room for the collars on the crank pin. The collars are on the eccentric itself. A special case is shown at H in Fig. 443; and where the motion is at right angles to that of the eccentric, the eccentric-face must be spherical. Both these arrangements make the full width of the face effective as a bearing. Note how the hole under the clamping-stud is arranged in Fig. 443, to make the surface continuous.

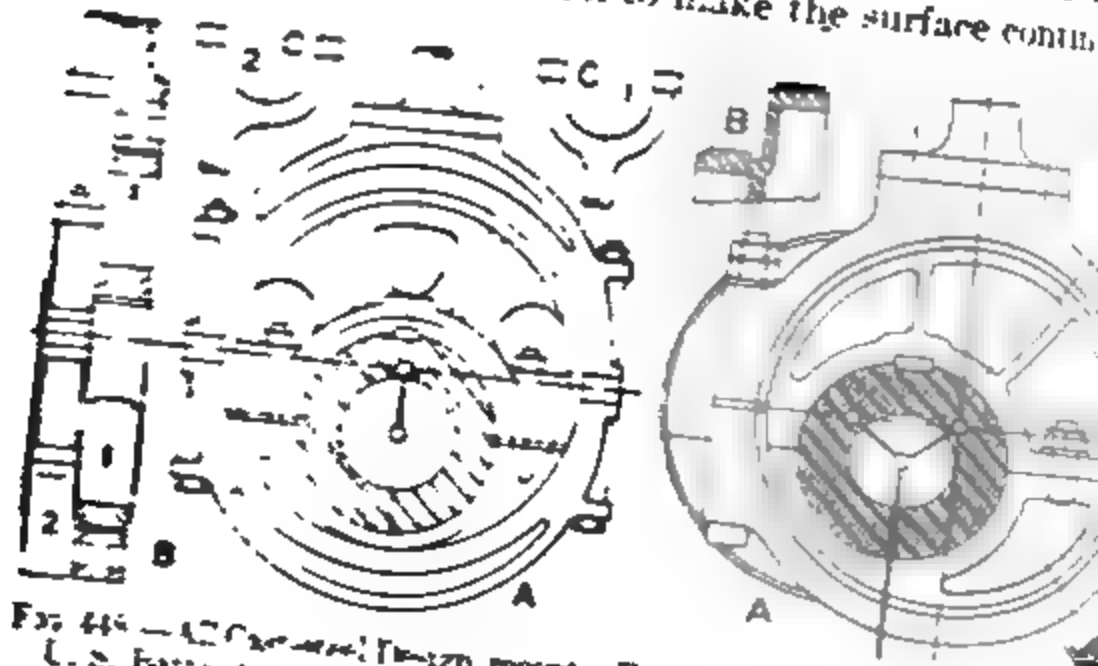


FIG. 448.—Cast-iron Eccentric Shaft recent U. S. Battleship. Scale 1 to 25.

FIG. 449.—Steel Eccentric Shaft with Forged Straps, U. S. Battleship. Scale 1 to 25.

**THE ECCENTRIC-STRAP.**—In general form this rod-bearing is analogous to the marine connecting-rod end in detail it varies as to form and material, manner of fastening the rod to it, construction at the joints, kind of rubbing-surfaces and provisions for lubrication. Cast iron is commonly used for stationary engines, typical cross-sections being shown at E and F in Fig. 443, and a rod example at C. Cast steel prevails where weight must be kept down, as in Figs. 444 and 448. For large

—speed marine engines, the straps are frequently of forged steel, as in Figs. 449 and 450, in which case simplicity of form is especially important; while small marine engines often have straps of cast iron or bronze, like Fig. 443 G.

**Rod-fastenings.**—A crude arrangement, used with a round rod in small engines, is to tap the hub on the strap and screw the rod into it, with a jam-nut. Far better is the connection in Fig. 444, which is easier to make and put together, and facilitating adjustment of rod-length. On locomotives, where there is considerable possibility of failure of lubrication at the eccentrics, the rods must be very stiff; hence the use of a deep rectangular rod, strongly reinforced in a stout nose on the strap, as in Fig. 444. Another type of bolted joint has the bolts in the plane of the eccentric and perpendicular to the rod-axis, a rather peculiar and special example being given in Fig. 451. The T-head fastening, very common in marine practice, needs no comment.

**Strap-joints.**—The simplest form is the plain flat joint in Fig. 444; but stiffness is more completely assured by the use of narrow straps, as in Fig. 443, because

there is always a possibility that a continuous flat surface may be just a little high in the middle. Very seldom is the joint recessed to make the halves match positively, because it is an easy matter to make the bolts act as wedge-pins. Usually there are shims or distance-blocks at the joints, and with forged straps these are sometimes quite long, as in Fig. 450.

A very special arrangement is shown in Fig. 451: at one side the halves are practically hinged together, at the other a stout stop-screw determines how closely they shall approach; so that exact adjustment can be made very quickly, and much more easily than by changing or filing down

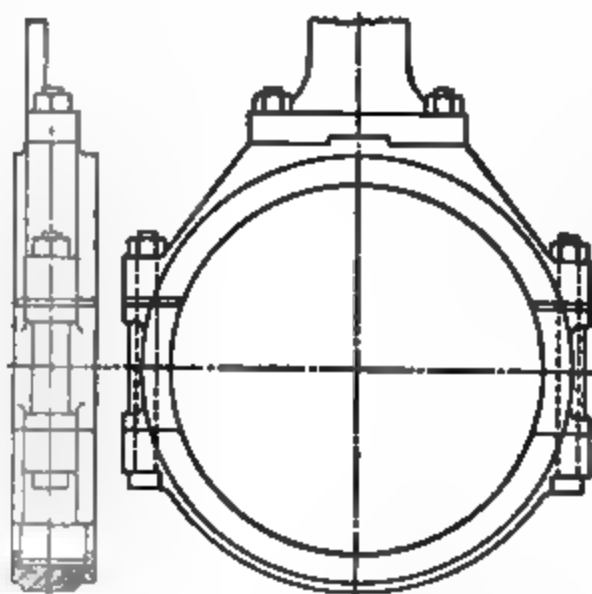


FIG. 450.—Forged Steel Strap for Large Cruiser. Scale about 1 to 27.

liners. Further, the compressive effect of the bolts is concentrated at the middle of the joint (on just the opposite principle to that in Fig. 443), with the idea of not distorting the strap by any possible force-action in the joint.

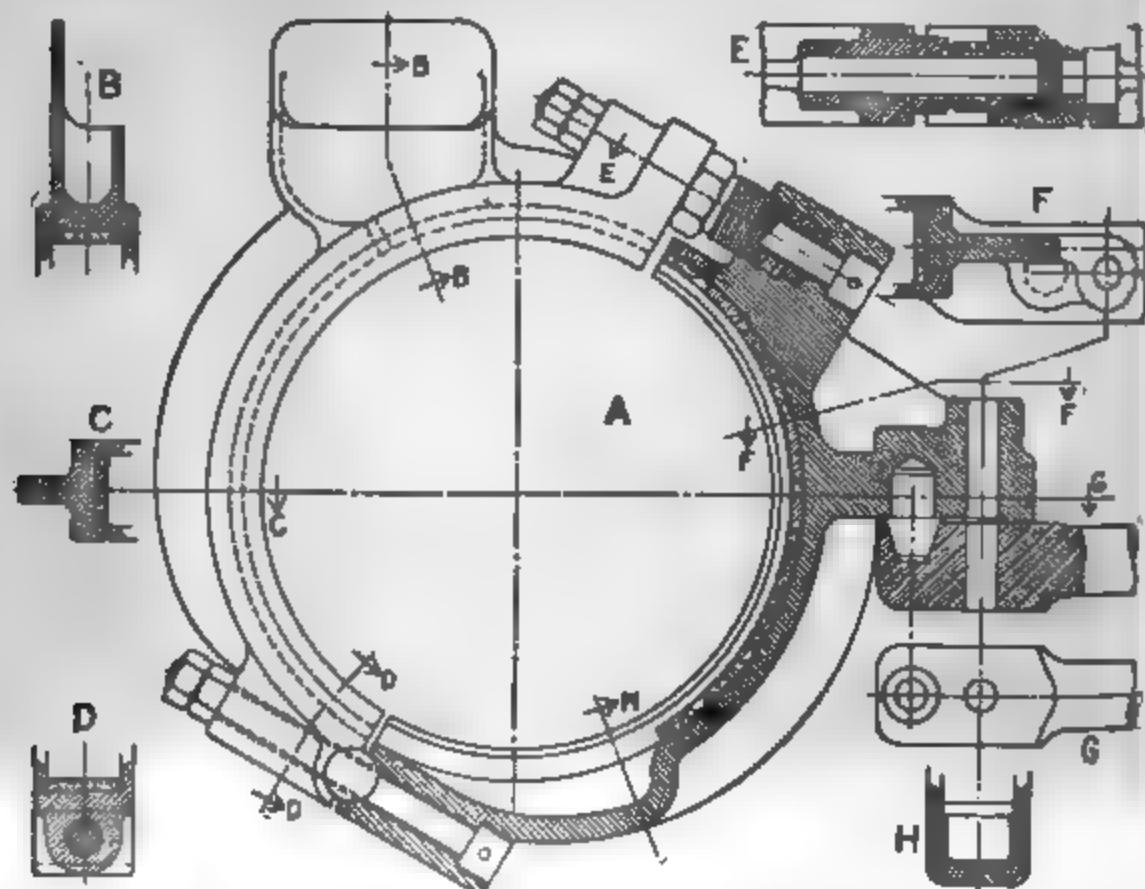


FIG. 451. -Adjustable Eccentric-strap from Engine in Fig. 205. Scale 1 to 6 for 12" x 12" Engine.

*Rubbing-surface and Lubrication.*—Not infrequently, a simple bearing of cast iron on cast iron is or has been used; but in all the better grades of practice the strap is lined with some anti-friction metal. In Fig. 444, tin is self-soldered upon the steel surface, just as on the cross-heads in Figs. 286 and 289; Figs. 443 A and G show the detail of babbitt linings cast in pockets; while separate shells fastened into the strap, are shown by Figs. 443 J and 451.

To supply oil in Fig. 443, an ordinary sight-feed drip-cup is screwed into the boss on the strap. Marine-engine straps, such as Figs. 448 to 450, have oil-pipes like those on the connecting-rods, coming from cups fastened well up on the eccentric-rods. On small high-speed stationary engines, oil is supplied from a

fixed cup or pipe, and received by some such device as the catch-cup in Fig. 451. In Figs. 444 and 451 we see pockets formed in the lower part of the strap, to hold a supply of oil. On Fig. 444, the boss at the upper left-hand corner is for a grease-cup, this lubricant remaining in reserve against overheating.

(c) VALVE-GEAR RODS.—The general form of the eccentric-rod and the other connecting-links in the valve-gear can be pretty clearly seen on a number of the pictures in the first part of Chapter VIII., particularly in Figs. 201, 208, 210, 212, 215, 217, and 221; as also on the general drawings of valve-gears at Figs. 366, 368, 391, 394, and 455.

On large engines, the eccentric-rod usually tapers from the strap toward the other end; and long connecting-rods, used in gears like the Corliss, often taper from the middle toward the ends. In the link-motion, especially on locomotives, the rectangular cross-section for the rod-body is very common; elsewhere, the round rod is almost universally used.

*Rod-ends.*—The joints in the locomotive valve-gear are nearly always of the solid, non-adjustable type, with case-hardened bushings on hardened pins; then whatever the form of the end, it is forged solid with the body of the rod—see Figs. 366, 367, and 455. The same type of bearing has been used on some stationary engines; but the great majority of designers have preferred arrangements which permit the taking up of wear—using therewith one of the harder bearing-metals, generally brass. All the types of connecting-rod ends have been adapted to the valve-gear: thus Fig. 452 I. shows the old-fashioned key-and-gib joint, while the bolted strap is seen on Fig. 217; the solid, mortised-out end like Fig. 304 is sometimes used; Fig. 368 has the half-solid marine end like Fig. 308; and Fig. 448 shows eccentric-rod forks shaped for square-box marine ends, as in Fig. 309. On stationary engines, however, the separate head, screwed on the rod and locked by a jam-nut, is by far the most common arrangement; this head either being solid, with a half-box set up by a screw as in Fig. 452 II. or by a wedge-key as in Fig. 460, or else having the "marine" form shown in Fig. 452 III.; usually the head is all of brass. When facility for rod-length adjustment is desired,

The ends of the rod are threaded right and left; and the nuts are made the same, held with a wrench.

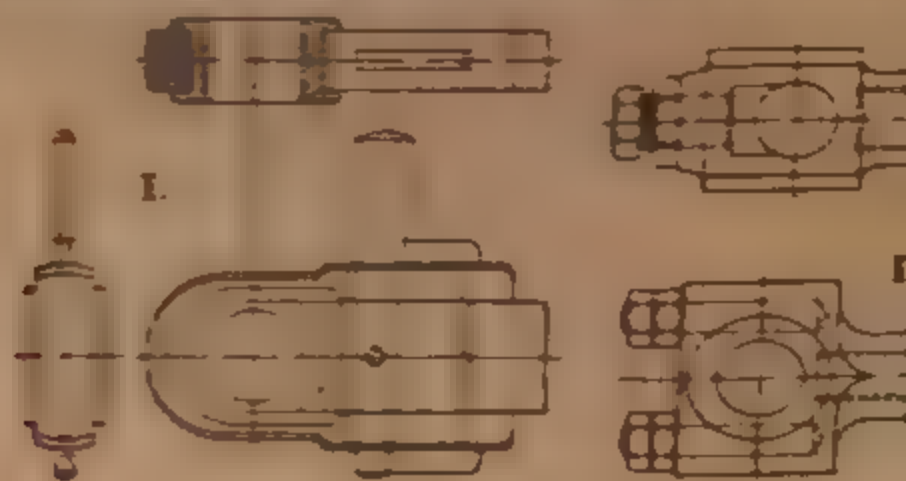


FIG. 45. — Valve-rod Ends.

The rod as in Fig. 460 or by forming a hexagon the end of the rod as in Fig. 301.

2. **ROCKERS AND VALVE-ROD SLIDES.** The name "rocker arm" is applied to any oscillating lever used for the purpose in a valve-gear. Absolutely none is seen in Fig. 212 where the short rockers at the corners serve merely to guide the rod-joints, without giving the motion transmitted. Very common in the type in Fig. 210 which gives the rod-motion the displacement in the plane of the mechanism, with magnitude usually an increase. Analogous to this lever is the crank-rocker on Fig. 208. The exhaust Fig. 215 are driven by a plain reversing-rocker, parallel to the motion. For a direct transfer from one plane the U-shaped device seen on Fig. 201 is most often employed. A lever in Figs. 2 and 4 serves the same end, guides the rod-joints by combining movements perpendicular to each other. Most complex of all is the reversing-rocker shown in Fig. 366, which transmits the motion and reverses the rod-movement.

The U-rocker in Fig. 201 is of the built-up type, cast iron arms keyed to the shaft, and stiffened by pins on the ends of a lighter shaft which runs from



other. In Fig. 453 the main rocker is of the simple keyed form, that for the cut-off valve is all of cast iron, except that it is clamped upon a hollow shaft in which the first shaft has its bearing. The pins at  $E_1$  and  $E_2$  are carried by the catch-blocks on the rods instead of by the rockers, these blocks being made on

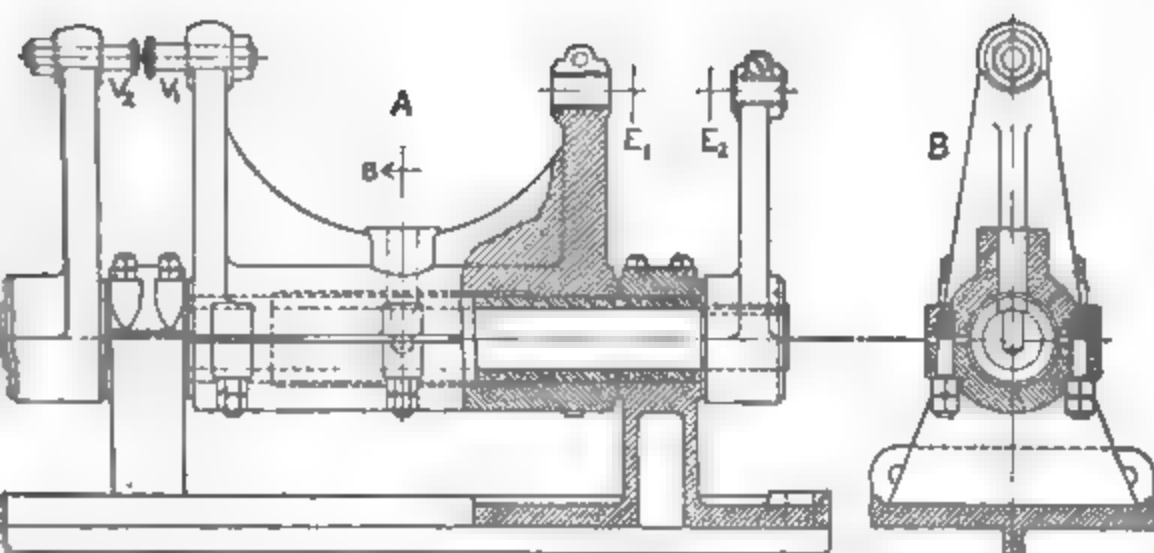


FIG. 453.—Double Rocker, to Drive the Valves in Fig. 440.

the principle of Fig. 483 III. Note the hole for the starting-bar, best shown in view B, so made that when the bar is inserted the two rockers are locked together—both valves being released from the control of the eccentrics and moved by hand in stopping and starting the engine.

*Valve-rod Slides.*—Quite often the valve-rod is coupled directly to the rocker, as in Figs. 201 and 366, its own flexibility and the small departure of the pin from a straight-line path permitting this. With more rigid construction, a plain knuckle-joint is sometimes formed just outside the stuffing-box, with nothing but its own stiffness, and perhaps a little support from the packing, to hold the valve-rod in line. A small hollow slide-block or cross-head, suitably guided, may be formed around this joint; but the solid block with a pin projecting from the side of it, as in Fig. 454, is more usual. An advantage of this arrangement is that it can give all the offset needed when the eccentric is close to the bearing, Fig. 206 presenting an example.

Fig. 454 I. belongs to the valve-gear in Fig. 455, A being a

front and B a top view; 1 is the rocker and 3 a valve, supplied with a key-and-socket joint as in Fig. 453. The rod is cast a propeller-shaped flange *N*, which is fastened to the main guide-yoke. On the right is shown a Meyer gear like that in Fig. 452, in the form of the guide-tracker. At III. and IV. are

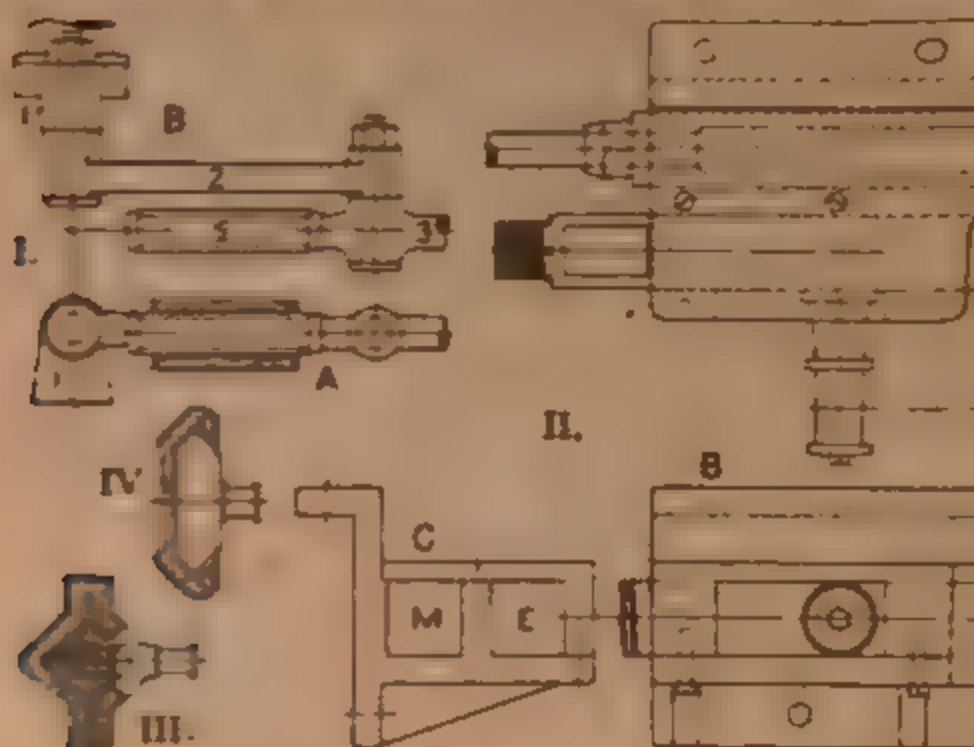


FIG. 454.—Valve-rod Slides.

of another type of block, which combines facility of adjustment for wear with security against turning of the block — holds the axis of the pin definitely horizontal; I is adjusted by means of liners in the joint, IV. by means of the little facing-strips, which are screwed up under the little facing-strips, which are screwed up to form a part of the rubbing-surface.

(c) DETAILS OF THE LINK-MOTION.—Fig. 455 is a diagram of a motive valve-gear which is a decided improvement on the American practice, in that the suspension-rods 10 and extension-rod 7 are made double, taking hold on the link. This gives simpler stresses and better insulation of the pressure upon the pins. In view of the eccentrics and of their rods is given to show the motion of the link and the two sets of suspension-links, 10 and 8, are

into the horizontal plane. Incidentally, this drawing shows a solid link, cut out of one piece, a very rigid form of the extension-rod which must get past the axle, and an unusual provision for lubrication. Not all of the oiling devices are here given, the rods 10 having cups fastened to them well toward the upper ends,

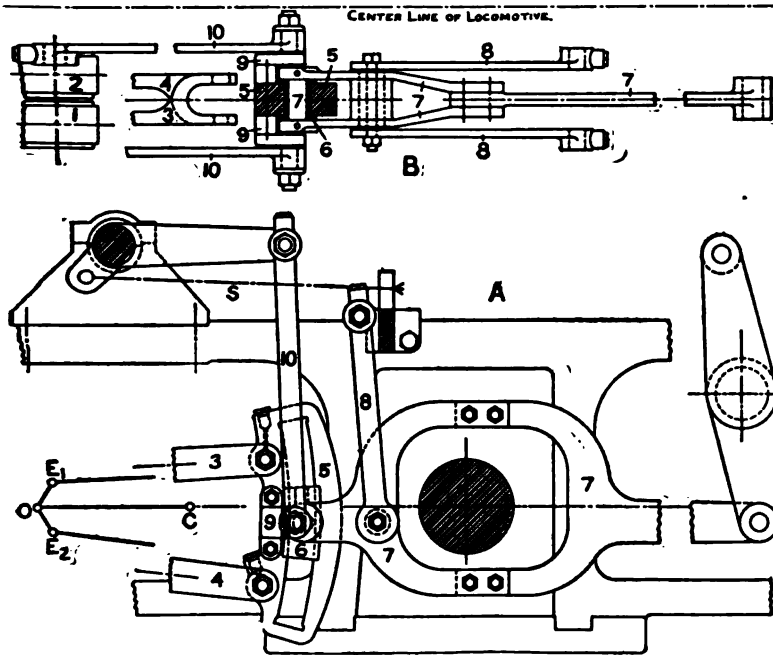


FIG. 455.—Locomotive Link-motion, Standard Penna. R.R. Design.

with pipes to the lower joints after the marine fashion; similarly, cups at the top of the loop in 7 supply the joint 7-8.

*The Box Link.*—Fig. 456 shows a link which is equivalent to the two-bar marine type—see Fig. 368—in having the rod-pins on the center-line and yet permitting the block to slide into the full-gear positions. View C shows how the halves of the link, with the end-blocks out, can be got into the solid jaws of the eccentric-rods, while D gives the form of the end-block. This enclosed form of link, seen in Figs. 391 and 394 also, is inherently

the most difficult to make, because two curved surfaces, required to be exactly alike, must be machined separately; while the two

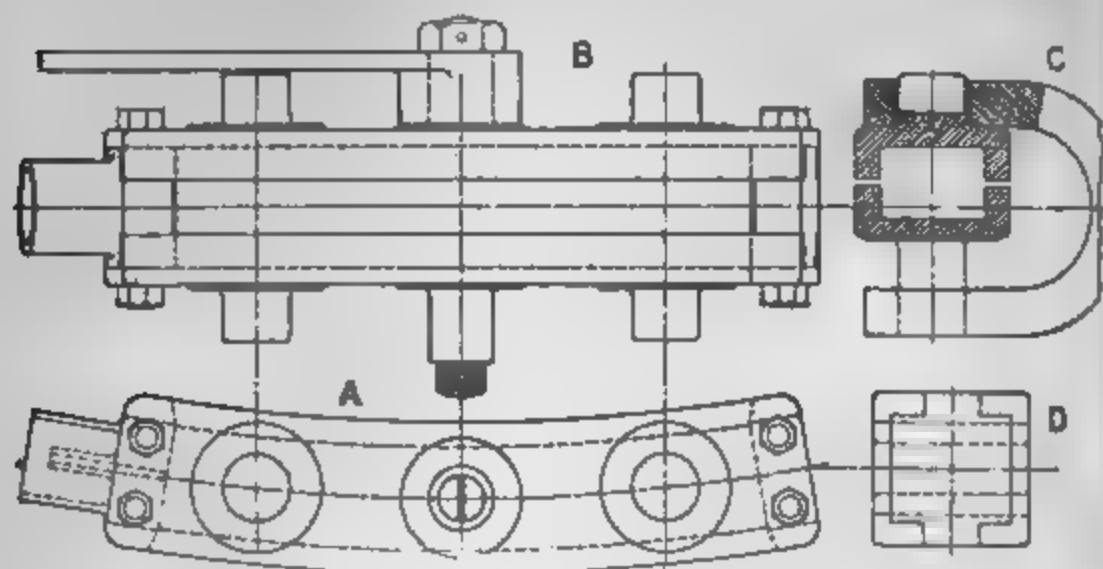


FIG. 456.—Enclosed or Box Link, from French Locomotive.

bars of a marine link can be clamped together and finished in the same operation. Sometimes the link is a solid curved bar, with

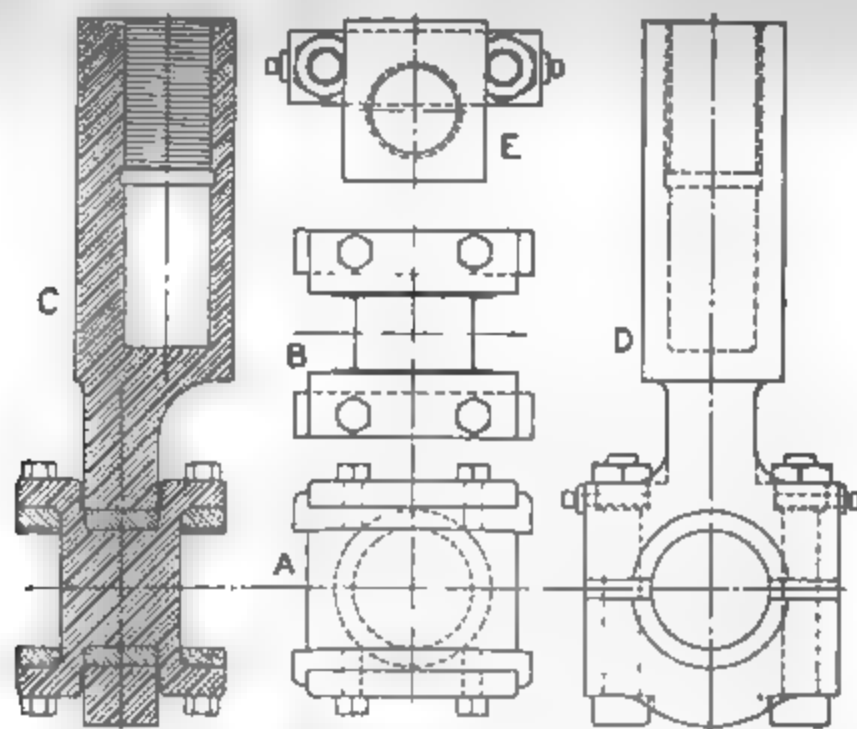


FIG. 457.—Block for Marine Link, with Valve-rod End.

the block surrounding it; but this is necessarily of the partial-gear type, like the one in Fig. 389 II.

*The Marine Link-block.*—A block like that in Fig. 368 is shown in detail by Fig. 457, together with a separate valve-rod slide; quite often, however, the valve-rod is continuous, an enlarged portion working in the guide-bracket, as in Fig. 368. Note how the body of the slide is drilled deeper than needed for the screwed joint, to diminish its weight. The block for Fig. 456 must be like this one in having the pin formed at the middle.

*Marine-engine Valve-rod Guides.*—From Figs. 233 and 234 can be got a good idea of the valve-rod guide used on marine engines, carried by an inverted A frame bolted beneath the cylinders. Especially strong guiding of the rod is necessary when two piston-valves are coupled to a wide cross-head, as in both the figures just referred to, for there is always the possibility that one valve will have a higher resistance than the other.

### § 56. The Corliss Valve-gear.

(a) A TYPICAL CORLISS GEAR, without special or novel features, is shown by Fig. 458, partly in skeleton outline; it is very much like that on the engine in Fig. 210, and belongs to the cylinder shown in Figs. 261 and 262, where the form of the valves may be seen. Besides the mechanism on the cylinder, we have here an outline of the crank-eccentric and of the rocker-arm, at A and B. Starting at the eccentric, we shall now consider in detail the form and action of this valve-gear.

(b) MOTION OF THE WRIST-PLATE.—By means of eccentric-rod 1, rocker-arm 2, and reach-rod or hook-rod 3, motion almost harmonic is given to the point H on the wrist-plate 4; and we may say that this piece oscillates in practically harmonic motion, conceiving its angular movement as determined by the linear movement of H. The reach-rod is not permanently connected to the wrist-plate, but merely hooks over the pin, so that the valves can be moved by hand, with "starting-bar" S, in starting and stopping the engine.

(c) NON-HARMONIC MOVEMENT OF THE VALVES.—Through the valve-rods 5 and 7 and the cranks 6 and 8 the valves are given an oscillation which is far from harmonic. Intentionally, the

opposite angular displacements of any valve-arm, from a mid-position determined by putting the eccentric at  $90^\circ$ , are made very unequal. This is clearly shown in Fig. 462, where the mechanism is drawn in this mid-position and the range of movement of each C-point is marked. The result sought and obtained is, that the valve shall have a wider and quicker movement in

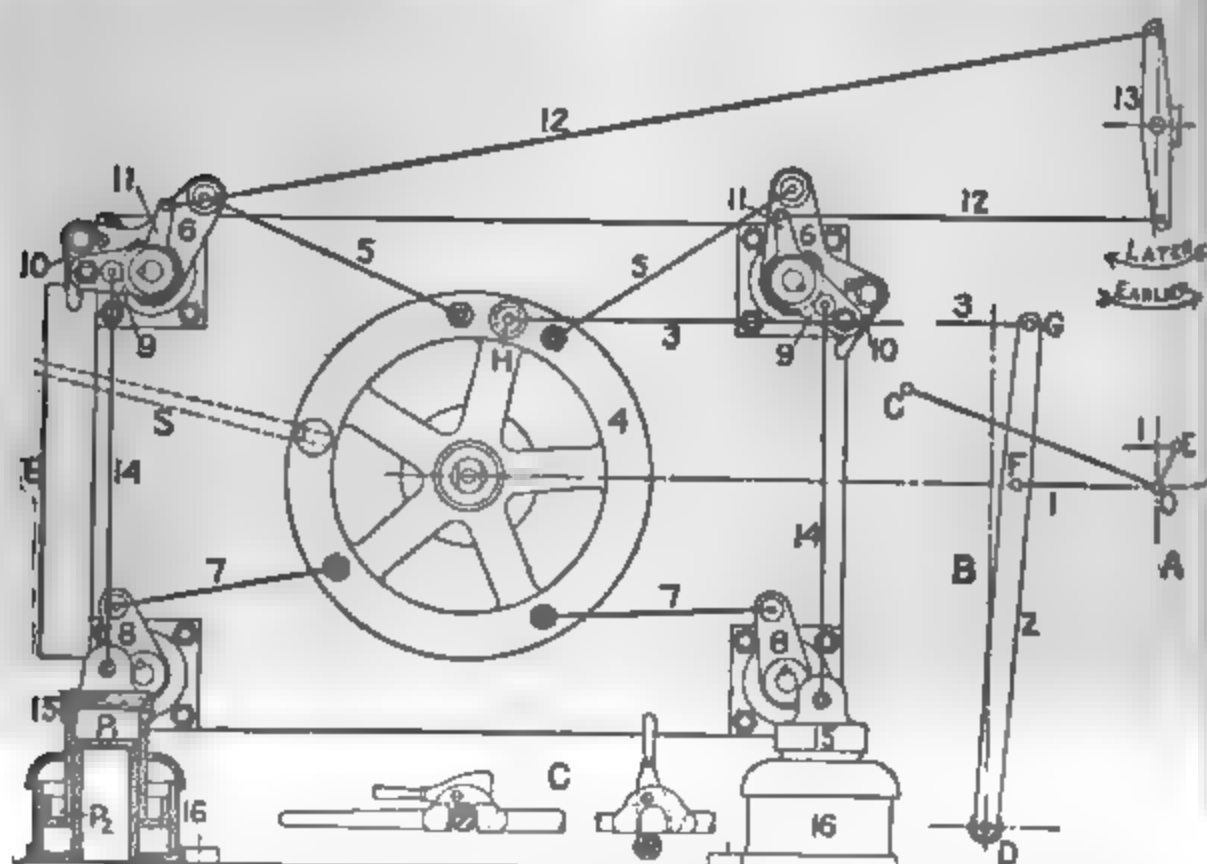


FIG. 458. —Corliss Valve-gear, Single-eccentric Type, with Full Wrist-plate.

- |                      |                          |                       |
|----------------------|--------------------------|-----------------------|
| 1. Eccentric-rod.    | 6. Oscillating cranks.   | 12. Governor-rods.    |
| 2. Rocker-arm.       | 7. Exhaust valve rods.   | 13. Governor rocker.  |
| 3. Reach-rod.        | 8. Exhaust-valve cranks. | 14. Dash-pot rod.     |
| 4. Wrist-plate.      | 9. Steam-valve cranks.   | 15. Dash-pot plunger. |
| 5. Steam-valve rods. | 10. Hook-claw.           | 16. Dash-pot body.    |
|                      | 11. Cam-ring.            |                       |

the direction for opening, a shorter and slower movement on the closure side.

The principles lying back of this valve-drive are illustrated in Fig. 459, which is an outline of the mechanism for the head-end exhaust-valve, taken from Fig. 462, and shown in mid-position and at the two extremes. This four-piece mechanism has for its moving parts the oscillating levers or cranks OD and VC and

connecting link or rod CD. If the two cranks in such a mechanism are to have similar and symmetrical movements, it must be arranged like that which consists of the rocker-arm, the reach-rod, and the wrist-plate (with the frame), on Fig. 458—the essential condition being, that when the driving crank is in mid-position, the other shall be parallel to it and the two perpendicular to the connecting-rod, or to a line of mean direction with which that rod remains very nearly parallel. With the wide departure from this condition seen in the Corliss valve-gear, the driven valve-crank has a motion very different from that of the driving wrist-plate.

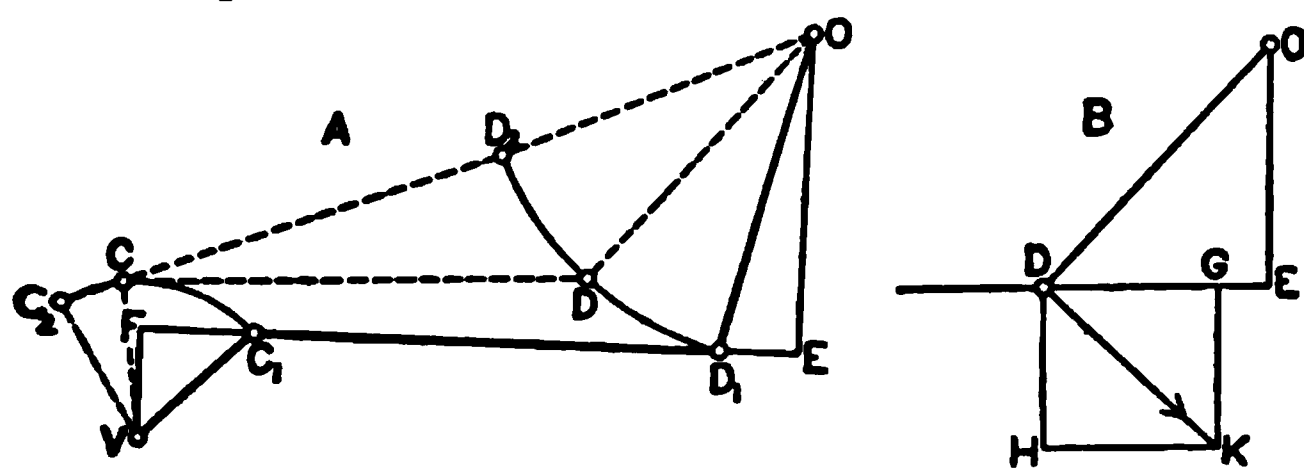


FIG. 459.—Outline of the Valve-mechanism.

A simple way to see how the velocities of these pieces compare at any instant is to note the property that if perpendiculars, as OE and VF, be dropped upon the center-line of the rod  $C_1D_1$ , an ideal mechanism OEFV can replace, for the instant, the actual linkage  $OD_1C_1V$ . With a given linear velocity transmitted along EF, the angular velocities of the cranks will be inversely as the radii OE and VF. Where OE is long and VF short, the valve turns rapidly; at the dead-point, just beyond  $OD_2C_2V$  in Fig. 459, where OD and DC are in line, the valve stands still.

That OE is the effective radius of the wrist-plate arm OD, to drive the rod DC, can be proven (if proof be needed) as at B in Fig. 459. Resolving the velocity DK of the point D into components along and perpendicular to the rod, we see that the transmitted component DG bears the same ratio to the radius OE that the total velocity DK does to OD: wherefore, a rod attached, for the instant, at E would be kinematically equivalent to the actual rod.

(d) **THE CUT-OFF MECHANISM** — The release end steam-valve is shown in detail by crank 9 is merely dotted in although the hook it is drawn in full. At B the several cranks are into one vertical plane, which involves a distance save overlapping the hook-claw or latch 10 by its center-line. All the valve-rods have heads and are adjustable in length by means of rings as are also the governor-rods 12 and the dash-pots.

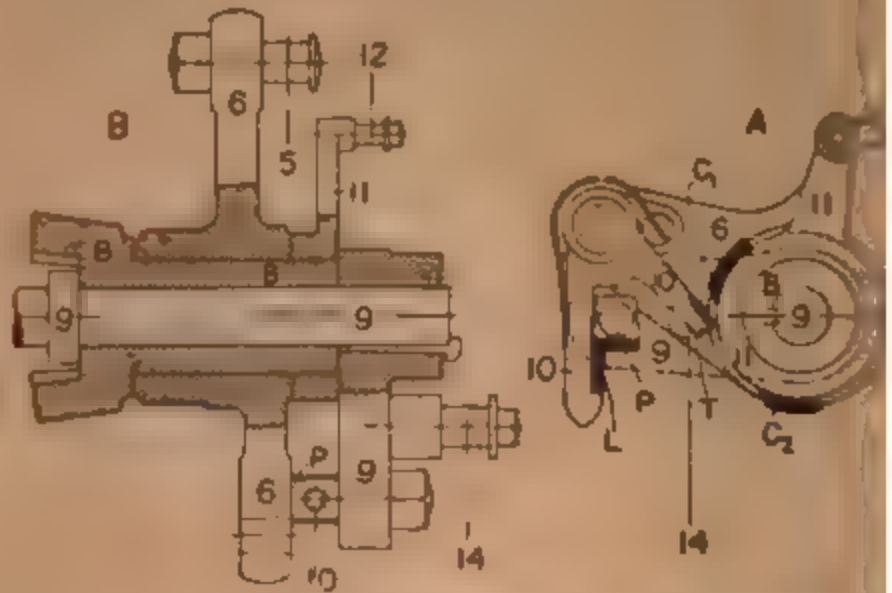


FIG. 400.—Detail of the Releasing Mechanism.

relative positions and the different motion-planes of the whole mechanism have already been illustrated.

The oscillating crank 6 has its bearing on the bonnet B. While it is turning toward the closed position from the previous cut-off, and at rest: as it travels, the latch L slides over the catch-plate until it is past the engaging point. As the crank 6 turns it turns the valve-stem with it, until it reaches the knock-off cam C<sub>1</sub> and L is pushed out of its position. The dash-pot pulls the valve quickly to the closed position, giving a sharp cut-off.

It is noted that all the hooking and tripping pieces are made of hardened steel, shaded in the drawing.



which can easily be re-adjusted or replaced—being held fast by small bolts or screws not fully shown on the drawing. A section of one dash-pot is given in Fig. 458; it is of the double type, the inner plunger  $P_1$  being mostly concerned with the vacuum-action, the annular plunger  $P_2$  with cushioning. A fuller description and discussion of the dash-pot will be found in Art. (p).

(e) CONTROL OF THE CUT-OFF.—The manner in which the time of cut-off is determined by the governor, through movement of the cam-ring 11, is apparent from Figs. 458 and 460. An important point is that the trip-arm must strike the cam  $C_1$  before the crank 6 gets to its limit of travel on the open side; otherwise the valve will not be released at all, and steam will be admitted through nearly the whole stroke. This means that the latest cut-off operation under control of the governor must begin a little while before the eccentric gets to its dead-point—rapidity of closure then depending upon the strength of the dash-pot. As will presently be shown, this places the limit of controlled cut-off at from 40 to 45 per cent. of the stroke in engines which have all their valves operated by one eccentric.

*The Governor.*—This is of the vertical fly-ball type for a Corliss engine, the one from the machine under consideration being shown in Fig. 461, where view A is taken from the cylinder, looking toward the shaft, while B is a view from the front of the engine—compare Fig. 210. The working of the governor-mechanism is self-evident, the sleeve 6 being moved up and down as the centrifugal force of the balls varies with reference to the downward pull of their own weight and of the balance-weight  $W$ ; and the running-speed can be changed by moving  $W$  in and out along its arm. The dash-pot 10, with a loose-fitting piston working in oil, is of the drag or damping type, and is put on to keep the governor from responding too freely to small and irregular impulses. The governor-rocker 8 is the piece numbered 13 on Fig. 458.

*Action of the Safety-cams.*—In Fig. 461 the whole mechanism is drawn in the starting position (for the engine standing idle), with the arm  $L$  resting upon the stop-ring  $S$ . This gives the latest possible cut-off, without release, the trip-arm  $T$ , Fig. 460, just working between cams  $C_1$  and  $C_2$ . The ring  $S$ , freely turned by

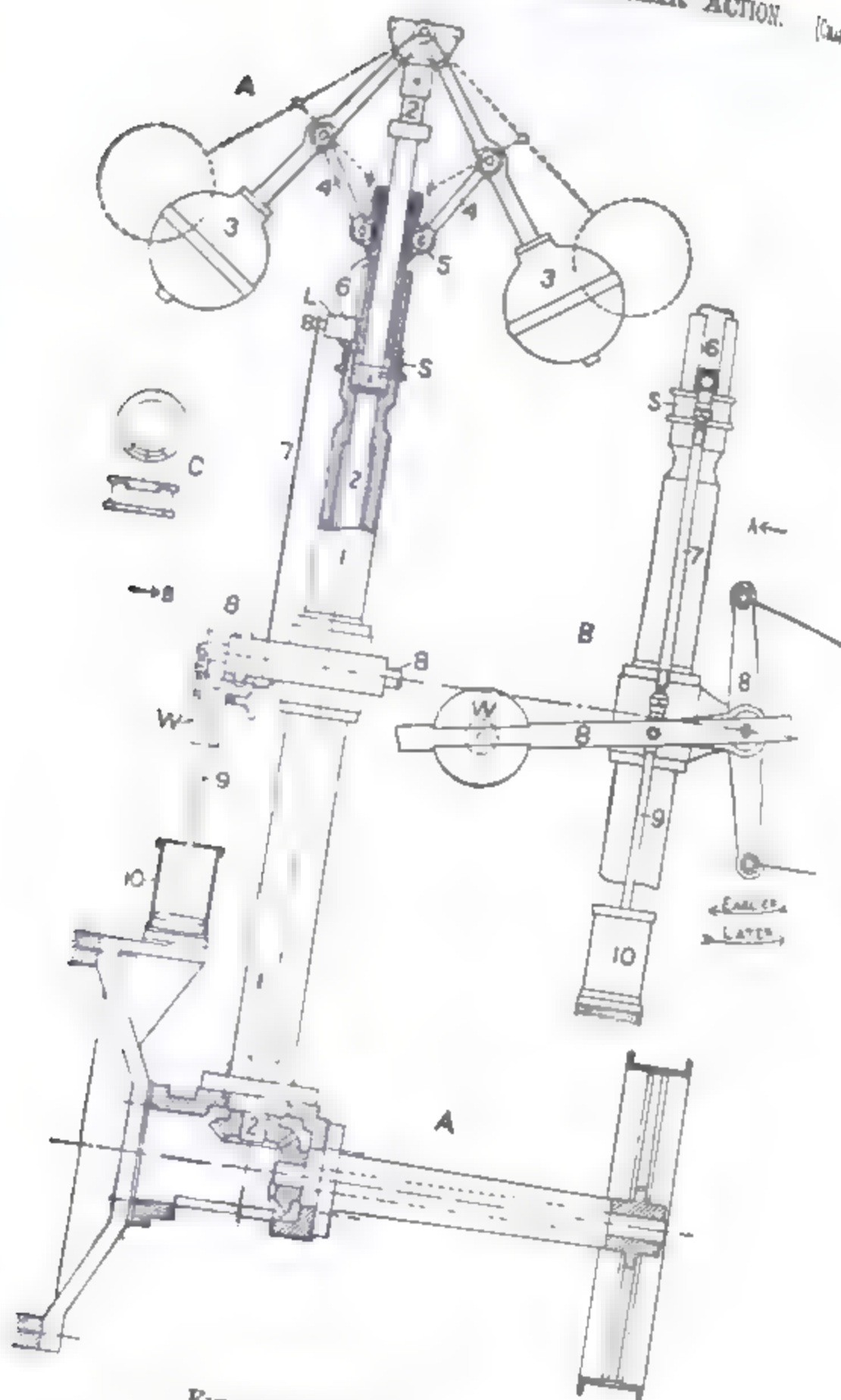


FIG. 461 — The Governor.

hand after the governor has lifted, has a part of its upper edge cut away, as shown at C on Fig. 461. When the engine is running, this notch is to be brought under L; then if through accident to its belt or for some other reason the governor ceases to turn, the balls will drop below the rest-position, the safety-cam  $C_2$  will be brought around so far as to prevent engagement of the valve-arm and opening of the valve, and the engine will be shut down. Further, when the governor is at its highest position the knock-off cam  $C_1$  should release the valve before it gets back to the point of admission, and thus again shut off steam so as to prevent a runaway. A disadvantage of this simple safety-stop is, that an overload sufficient to slow down the engine to any considerable degree will bring it to a dead stop, unless the engineer can foresee this condition and turn the ring to the normal rest-position; wherefore he is likely to leave it in this position, thus depriving the engine of protection against the effects of an accident to the governor.

(f) THE MOVEMENT OF THE VALVES.—This is best shown by plotting diagrams like Fig. 337, with the linear movement of a point on the valve-face profile, along its circular path, as ordinate and the developed eccentric-circle as base. For the steam-valves we at first disregard the releasing-gear, either imagining the oscillating crank to be fast to the valve-stem, or considering that we determine the movement of a point on this crank projected out from the valve-face. The dimensions necessary for a layout of the valve-gear are shown on Fig. 462: this particular example was measured up from the actual engine, and it will be noted that the valve-rods are adjusted to different lengths.

Having the mid-position outline drawn, we next strike the circle on AB, with a radius equal to that of the eccentric reduced to the upper end of the rocker-arm, or multiplied by the ratio  $GD/FD$  from Fig. 458. The eccentric-rod is so long that we may very well disregard its angular swing, and assume that the horizontal movement of H is harmonic. Then an angle-scale for the movement of the wrist-plate is got by dividing the eccentric-circle and projecting the points of division to the path of H. For clearness of drawing, this path is moved down to MN, on an arc

struck from  $O'$  with a radius equal to  $OH$ . It would, of course, be a simple matter to use the exact method of Fig. 357 in laying off the position-scale for  $H$ , but the gain in accuracy would be insignificant.

The four driving-points  $D_1, D_2, D_3, D_4$ , being all on the same circle with  $H$ , this travel-scale  $MN$  is next centered on each one: where

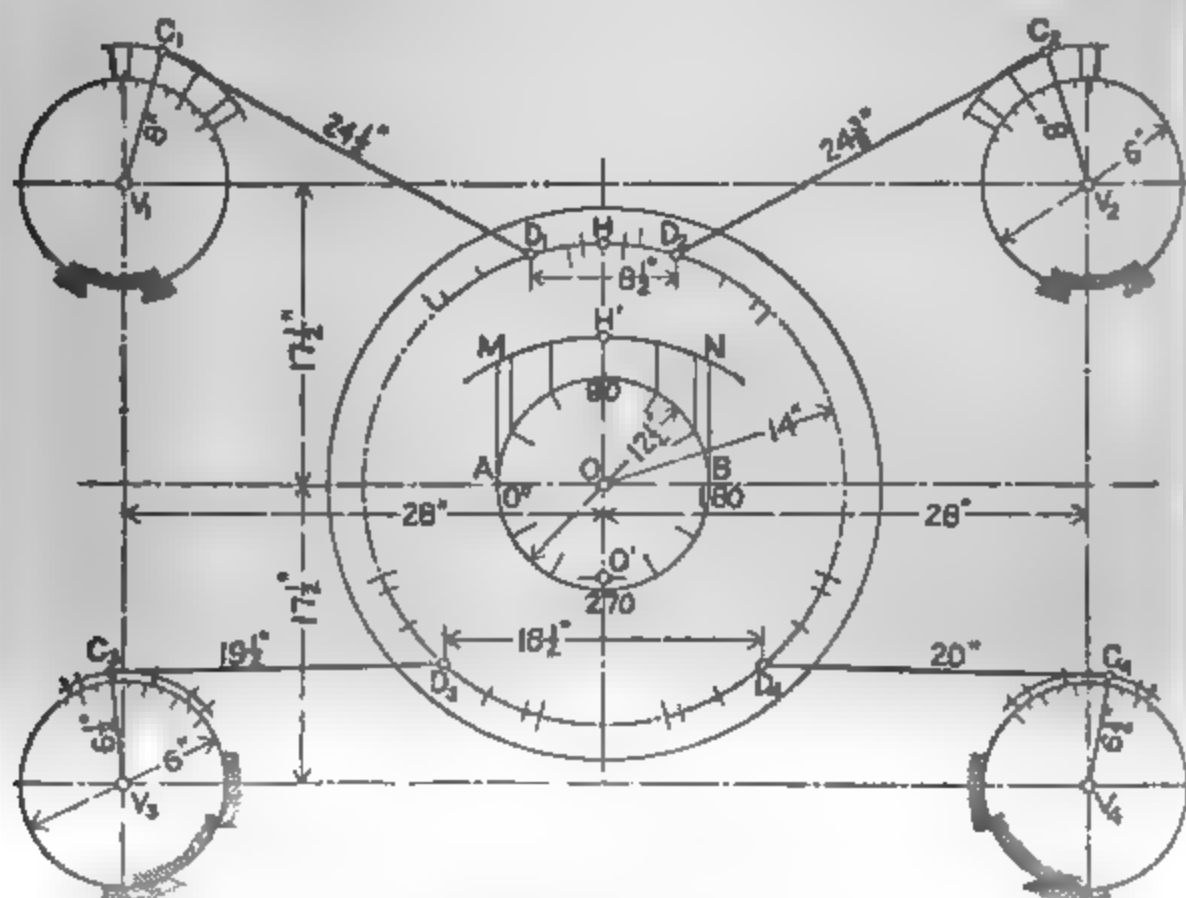


FIG. 462.—Diagram of the Valve-gear.

two paths overlap, as is the case with  $D_1$  and  $D_3$ , one scale is marked outside, the other inside, of the circle. The positions of the  $C$ -points are now found by striking off the rod-lengths, and are then projected radially upon the circles representing the profile of the full cross-section of the valve.

Fig. 462 is the picture of a drawing in which the mechanism as a whole was laid off half-size, but the valves drawn full-size. then the actual travel of the valve-surface is given by the operation last described, and it is only necessary to rectify the curved path in order to have the desired ordinates of travel. For the purpose a scale of inches laid off on the valve-seat, here mark

on the inside of each circle, and used in connection with the horizontal ruling on Fig. 463, is most convenient.

The movement-curves got by plotting the several sets of ordinates are given in Fig. 463, I. and II. for the steam-valves, III. and IV. for the exhaust-valves. The angle-scale at the top of the diagram is the same as that on the circle AB in Fig. 462. Travel in the direction which opens the port is represented by an upward ordinate for the head-end curves I. and III., by a downward ordinate for II. and IV. The distortion from harmonic motion is very clearly shown by these curves.

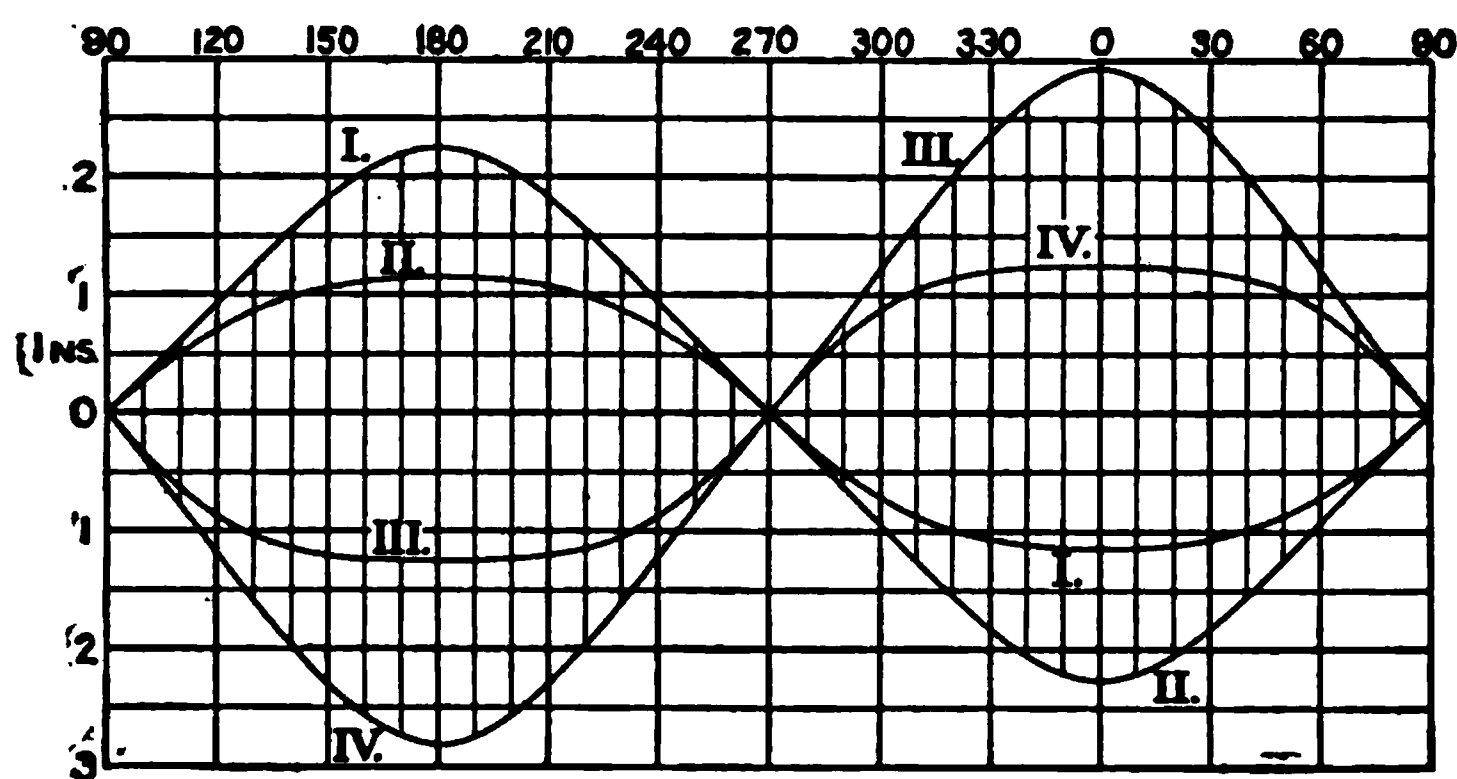


FIG. 463.—Valve-movement Curves.

(g) ECCENTRIC-SETTING AND VALVE-ACTION.—Having drawn the simple curves of valve-displacement in terms of eccentric-position, our next step is to combine with these the lap of the valves—measured in the usual way, in the position shown on Fig. 462—and thus find out the proper setting of the eccentric with reference to the crank. For present purposes it is enough to consider one end only of the cylinder, wherefore curves I. and III. are reproduced in Fig. 464. It is the operation of exhaust that determines the eccentric-setting, because both ends of this period have to be considered, as against only the beginning of the period of admission. Usually the lead for release, the angle TOB on Fig. 334, is less than the angle of compression, so that a small positive lap on the exhaust-valve is necessary: this is

represented by the distance of the line TS above the base-line, being here one-quarter of an inch. Since the lap must be less than that for release, the steam-lap and QR is drawn at three-eighths above the base-line, and to locate suitably the dead-center positions of the crank end, B for crank end, and find the angle  $\delta$  to be the angle of advance about  $16^\circ$ —by noting that when the crank is at A or zero, the eccentric is at  $106^\circ$  on its scale above the base-line.

*The Cut-off Action.*—Instead of following the action represented by the whole of curve I., the steam-valve can be closed according to the full-line curves 1, 2, 3, or 4, of which

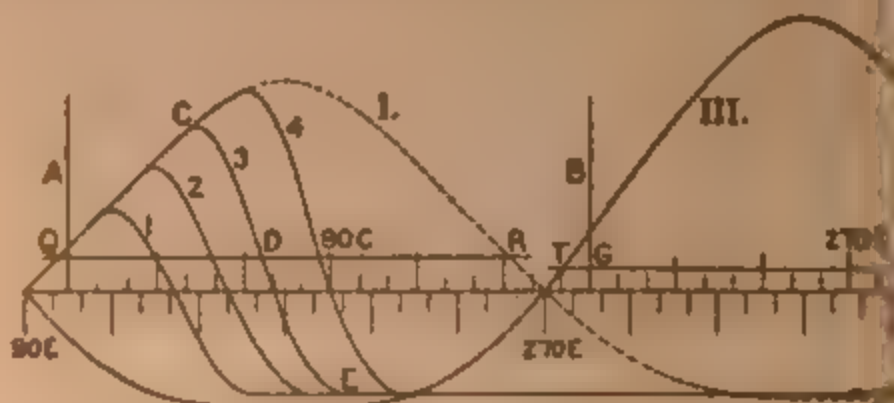


FIG. 464 Diagrams for the Head-end Valve.

CDE shows quick closure under the pull of the dash-pot, while the straight line EF the period of rest while waiting for admission. These curves are merely sketched in for considerations, since a mathematical determination of the dash-pot, while not incapable of giving results, would be, in length and complication, rather a small portion to the importance of the subject. Only an approximate calculation would be made in working out the results, and adapt the results of experience to the particular case.

*Valve and Indicator Diagrams.*—The steam-valve diagrams in Fig. 464 are re-drawn in Fig. 465, upon the steam-piston as base, the piston positions being found from Table VIII and including the effect of the connecting-rod. These are to be compared the autographic diagrams.

which were taken from this same engine, being drawn by a pencil moved by the drop-rod and working on the drum of an indicator

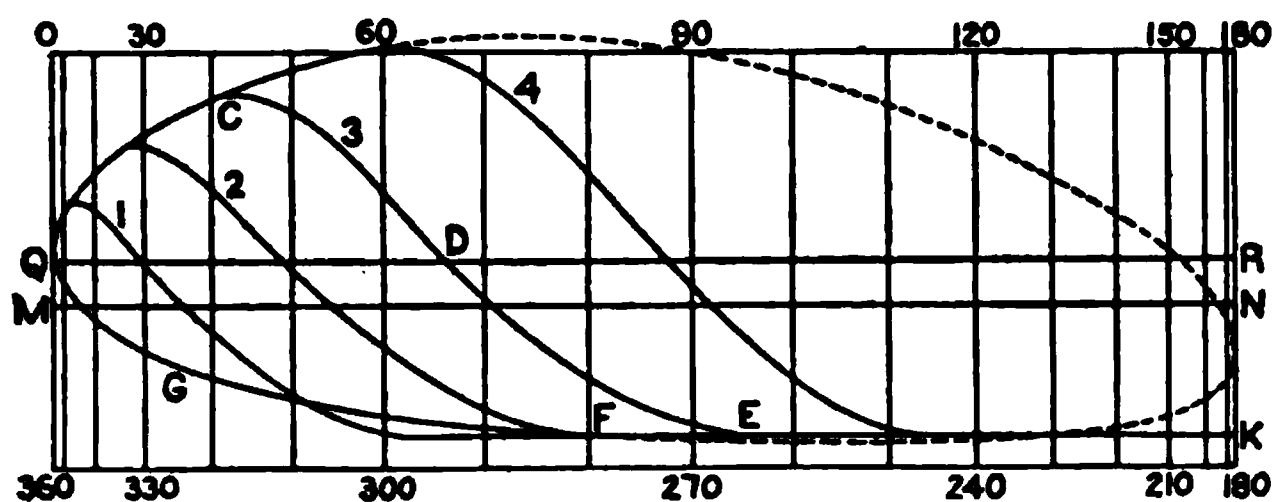


FIG. 465.—Stroke-line Diagrams for the Steam-valve.

which was connected to the ordinary reducing-motion from the cross-head. Simultaneous indicator diagrams are given at II. For the head end, the two types of diagram agree very well in proportions, the short vertical lines showing where cut-off takes place. These are first located on II., then transferred to I. with due regard to the difference in length of the diagram. The crank-end diagrams were evidently not really taken at the same time.

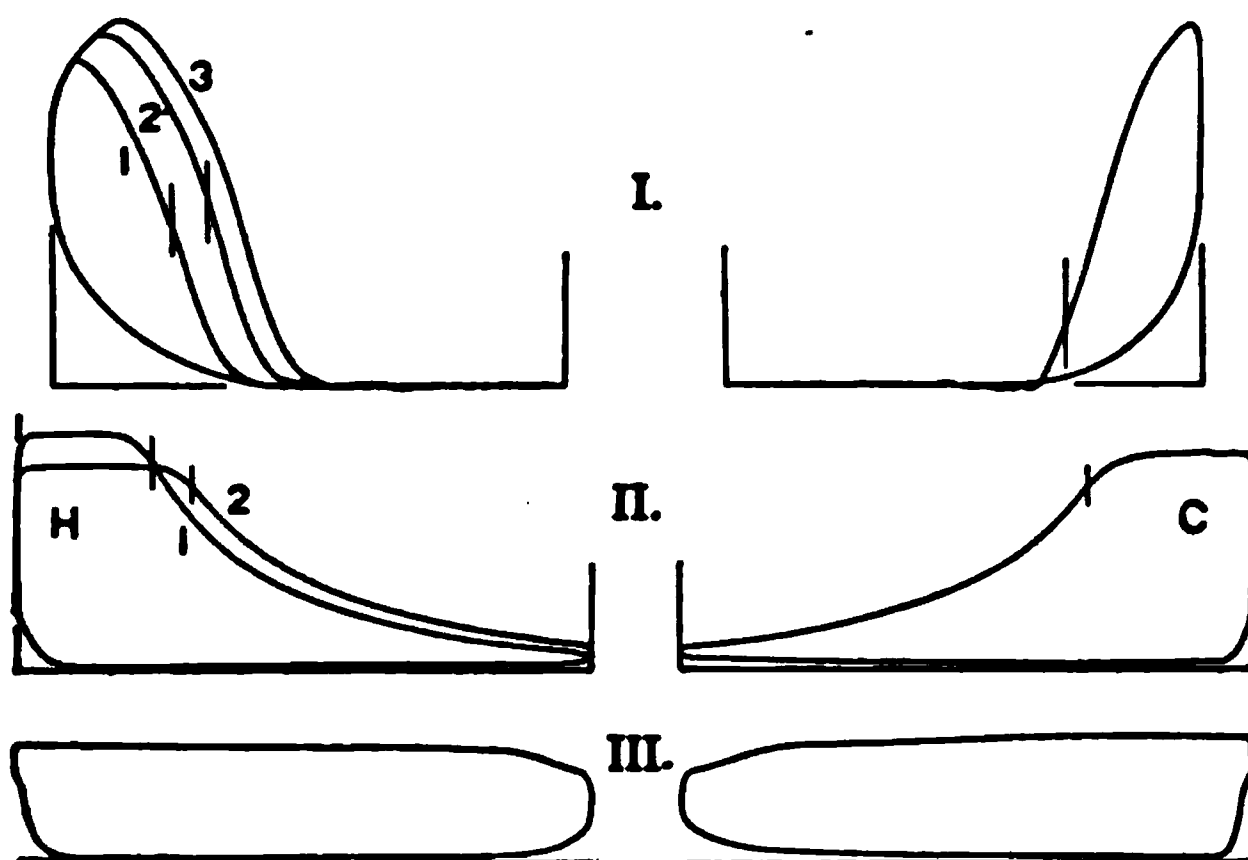


FIG. 466.—Indicator Diagrams from the Engine in Fig. 458.

The characteristic Corliss-engine diagram is well represented by Fig. 466 II.: the distinguishing features are, a horizontal



admission-line, a sharp and usually rather deferred till near the end of the stroke, and which begins late but rises rapidly on account of the pressure. At III. is shown what happens when the speed drops so low that the engine cannot keep up its load, and the releasing-gear fails to act—the period per revolution being now much greater than when working properly.



FIG. 467.—Faults in the Eccentric.

A great deal of the valve-setting to be done consists simply in adjusting the lengths of the eccentrics. Fig. 467, however, shows faults in the setting of the eccentrics. At I. it is seen that the valve needs to be advanced, at II. we see the effect of advancing it too far, apparent especially in the excessive outward slant of the rising admission-lines.

(A) VALVE-RESISTANCE IN THE CORLISS GEAR. The amount of work that must be expended in moving a valve is small, not so much because the valves are light as because they are very hard upon their seats as because the mechanical pressures are small. The steam-valve is opened while being opened; but it need not be closed before it is once open, any single-function valve is but to move easily by the substantial equalizing pressure around it. For the exhaust-valve especially, the gain in a marked distortion from harmonic motion. In Fig. 464 we see that the period of rest and of this valve coincides with the time when the pressure in the cylinder is high: and the wide and rapid



place after the pressure has been lowered by expansion and while the port is open.

Since the Corliss valves are, for a part of each revolution, definitely unbalanced, we have a better and surer basis for design of the gear, as to strength, than with balanced slide-valves which work under less determinable conditions.

(i) **VARIOUS FORMS OF THE CORLISS VALVE.**—As to the steam-valve, we first of all draw a distinction between inside and outside admission. The former is shown in Figs. 262 and 462 and in Fig. 468 II. and IV., the latter in Fig. 468 I. and III. With a valve formed as in Fig. 262, outside admission has the small disadvantage that the steam-current must follow a longer and more tortuous path; and the valve in Fig. 468 I. is an improvement in this respect.

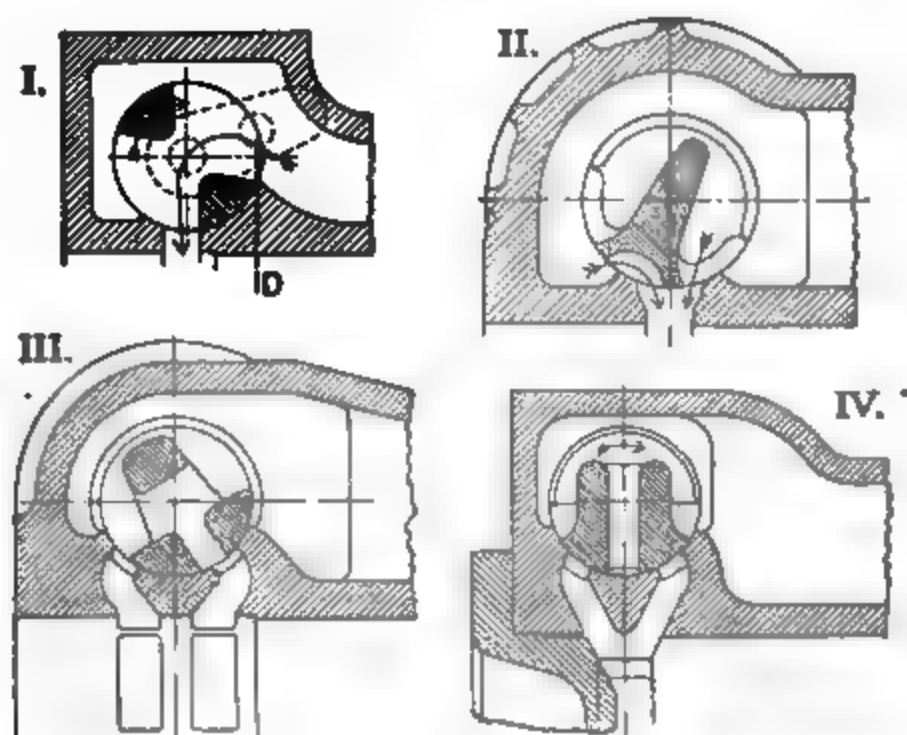


FIG. 468.—Various Steam-valves: I. Passage through Valve; II. Murray Valve; III. Brown Valve; IV. Reynolds Valve.

Of more importance is the matter of securing double admission. Fig. 468 II. shows how this can be done with a single port, using the principle of the B valve for the second opening. Note that after the right-side valve-face gets to the middle of the port—just a little beyond the position in the drawing—there will be

no further increase in the effective opening and that the mouth of the port is made wide of the passage by the width of the narrow



FIG. 469.—Detail of Cylinder-casting, Valve.

Two valves which may properly be called of interesting differences in form, are given in Fig. 470. The valve and steam-chamber belonging to the drawing in Figs. 261 and 262, and illustrates a form of steam-chamber, most clearly shown, horizontal section at D. Note how the heavy in the port are reinforced by stay-bolts.

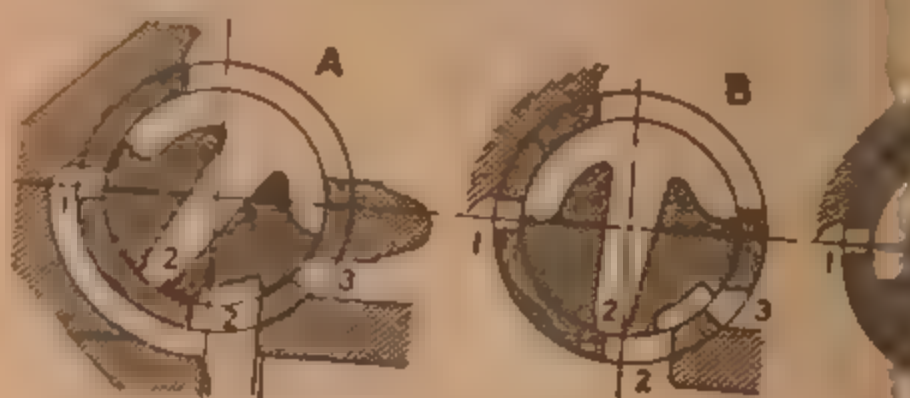


FIG. 470.—Fleming Triple-ported Valve.

A triple-ported valve, belonging to the engine in the valve-gear in Figs. 400 to 493, is shown in section in Fig. 470. View A is mid-position, with the three openings on the valve; in view B the third opening, through

greatest effective value; thereafter it is gradually choked, and is entirely closed for some little time before the extreme position at C is reached. This valve is controlled by a shaft-vernor, and it is for the short travels that go with early cut-off at the third port is intended to be effective—compare Figs. 351 and 352.

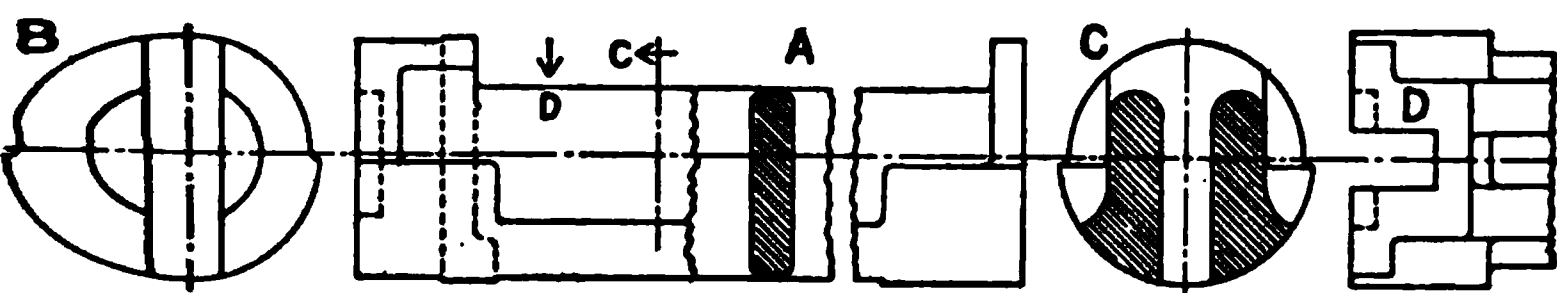


FIG. 471.—Detail of Reynolds Valve; Allis-Chalmers Co., Fig. 468 IV.

The detail drawing in Fig. 471 is intended to illustrate especially the construction of the valve at the ends or heads, and to show the amount of bearing-surface which these heads carry. Similar proportions appear at II. and III. in Fig. 468; but sometimes the heads are fully cylindrical, as on the valve in Fig. 470, which has also a full bearing in the two vertical webs that tie together the walls of the steam-chamber.

The exhaust-valve shown in Fig. 472 I. differs from that in Fig. 262 only in having a part of the cylindrical surface cut back, so as to leave merely a face wide enough to cover the port properly when closed, with sufficient lap over both edges. At II. the steam passes through the valve, and openings at top and bottom are closed at the same time. The especial object of this arrangement is to diminish cylinder clearance: but since it is hardly possible to make and keep the valve tight enough to prevent leakage at the top, this object can be only partially attained. The action of the valves at III., IV., and V. is obvious: the last shows a very effective way of reducing clearance to a minimum, by moving the valve partly into the cylinder-space. The critical position as to the possible interference of valve and piston is shown in the drawing, for reference to Fig. 464 shows that the exhaust-valve reaches its extreme position for closure while the piston is yet near the beginning of the stroke. The valves in Fig. 472 VI. call for no comment. A fact to be noted is that all these exhaust-

valves open by turning in the same direction, the working ed having the same movement—outward to open, inward to close, as in the case of a common direct D valve.

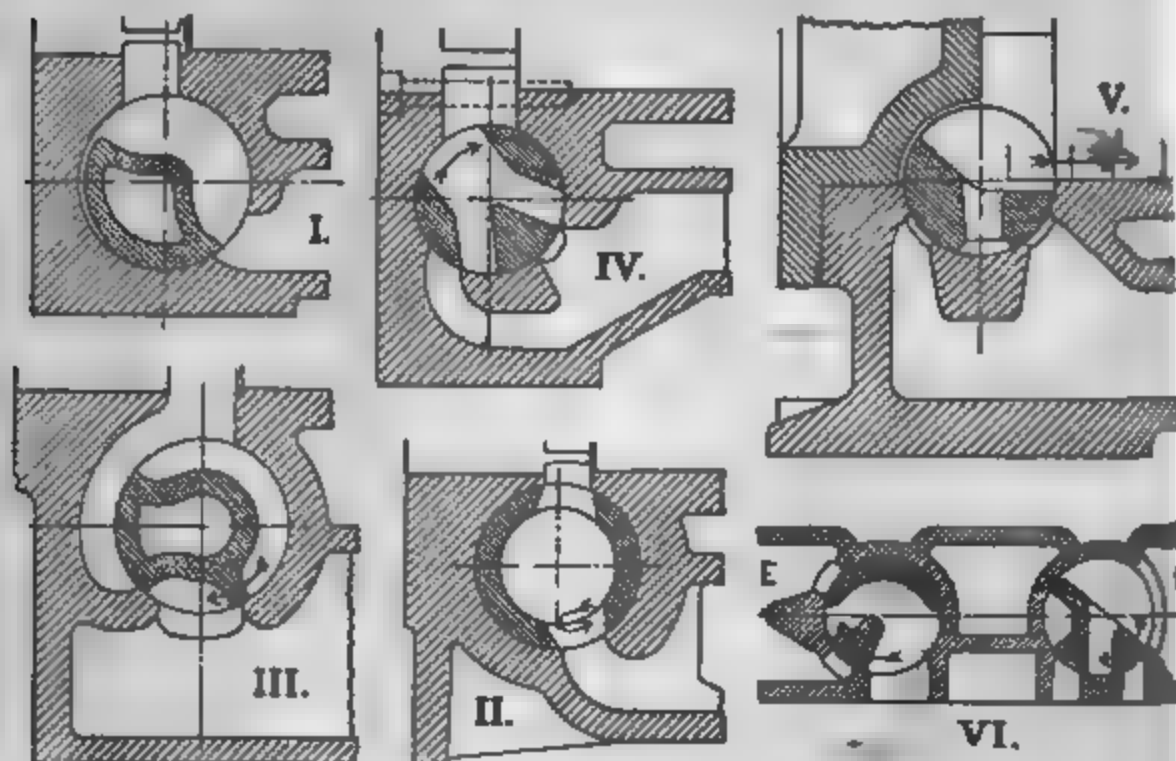


FIG. 472.—Various Exhaust-valves: I. Plain Single-ported Valve; II. Fleming Valve; III. Murra Valve; IV. Reynolds Valve; V. Brown Valve; VI. Valves in Head, Cylinder in Fig. 268.

(j) VARIOUS FORMS OF THE CORLISS GEAR.—Some important general relations are illustrated in Fig. 473. Two basal ideas are, first, that so much of the wrist-plate as serves to drive any one valve (any HOD on Fig. 462 or Fig. 473) may be considered as simply a rocker-arm, either direct-driving or reversing; second, that the valve-crank drives directly when, as at the steam-valve in Fig. 473 I., the point K and the working valve-edge move in the same absolute direction; but it is indirect when the driving-point K is opposite to the valve-edge, with respect to the axis of the valve. Then for the normal wrist-plate drive, with H at the top as in Fig. 458, the outside-admission valve is of the "direct" type (compare Fig. 336) and is directly driven; with inside admission, reversal of valve-movement is compensated by the change to an indirect valve-crank; and the exhaust-valves

selves direct, are driven by two reversing-levers which neutralize each other; wherefore the normal eccentric-setting is "direct", as indicated at COE. Only by connecting at the bottom, or by the use of a reversing-rocker between the wrist-plate

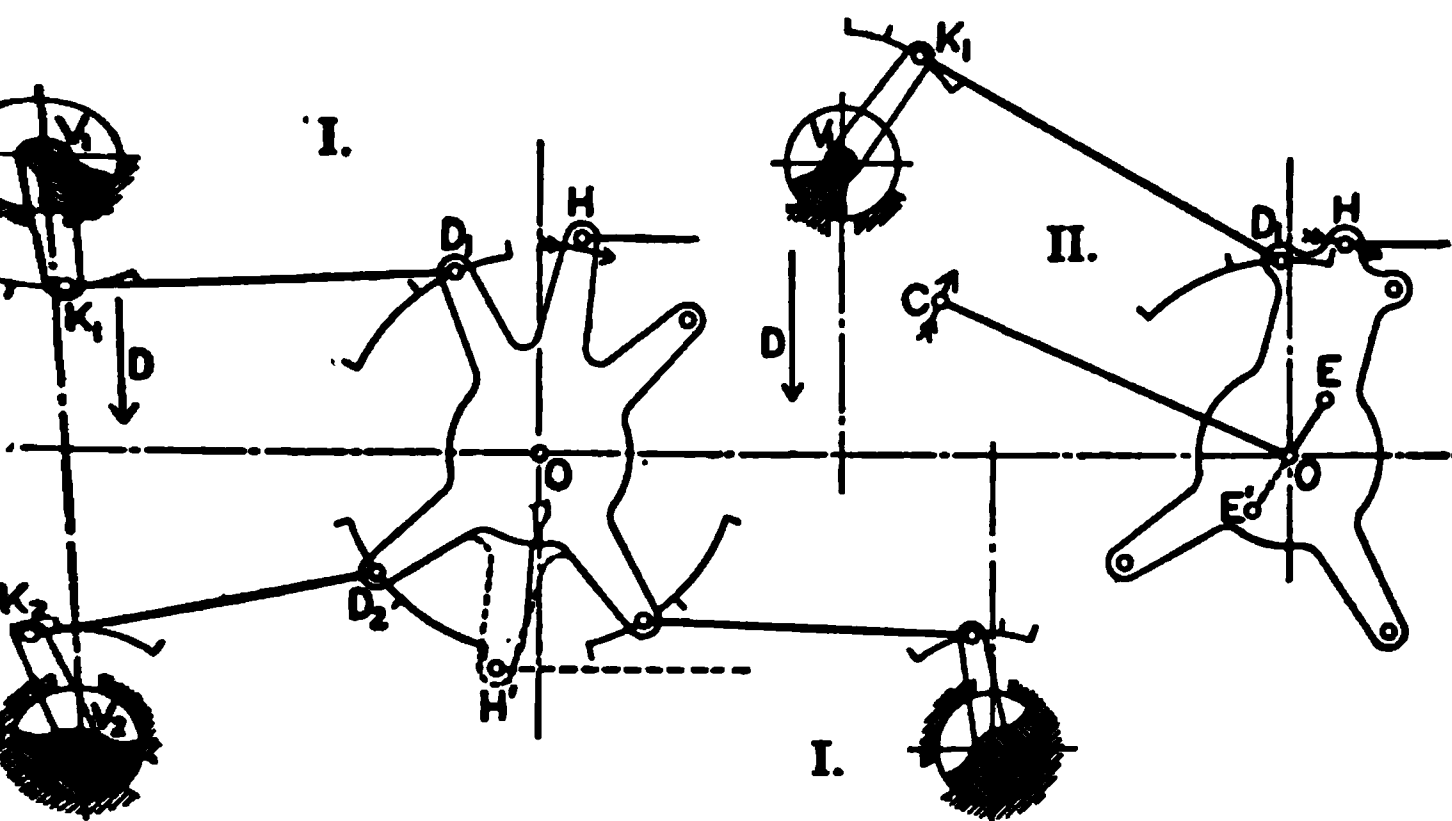


Fig. 473.—Typical Arrangements of the Common Corliss Gear: I. Outside Admission; II. Inside Admission.

and the eccentric, is the latter thrown into the indirect position E'.

A little study of Fig. 473 will show that if the desired distortion from harmonic motion is to be secured, and the several valve-rods kept clear from crossing and interfering with each other, the only possible ways of connecting the valve-arms to the wrist-plate are those shown on that figure. In every case, the rod DK is in tension when opening the valve, or the valves are pulled open and pushed shut. Another simple relation is, that the outside-admission valve has the drop-rod inside, and *vice versa*.

Fig. 474, besides showing the general arrangement for this case, is intended to bring out the particular point that on a vertical engine the two drop-rods must be on opposite sides of their respective valve-spindles—with corresponding differences in the arrangement of the two releasing-gears.

One important step in the development of a Corliss gear

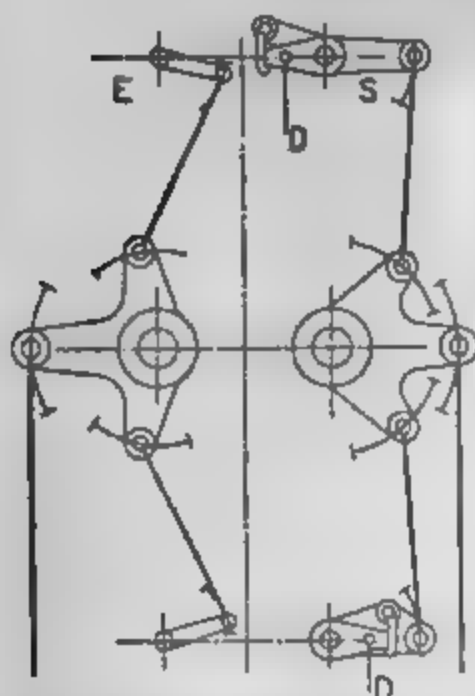


FIG. 474.—Corliss Gear in a Vertical Engine, with Two Eccentrics, and Valves in Cylinder-heads.

for higher speed is the adoption of the double-ported valve, which diminishes the amount of movement required; another is to reduce the mass of the valve-gear part especially by making the wrist-plate as light as possible, or by getting rid of it altogether. For the steam valves there is no great advantage in non-harmonic motion, with a releasing-gear; so that they can very properly be driven by a simple and direct system of rods, as in both cases of Fig. 475. But for the exhaust valves the wrist-plate effect is highly desirable, as brought out in Art. (k) so that I. shows a better design than II. in Fig. 475. Another example of the use of a separate bell-crank near each

valve, instead of a central wrist-plate, is given in Fig. 490. In this connection it may be well to remark that a number of builders offer to furnish Corliss engines to run at speeds as high as 150 or 160 R.P.M.

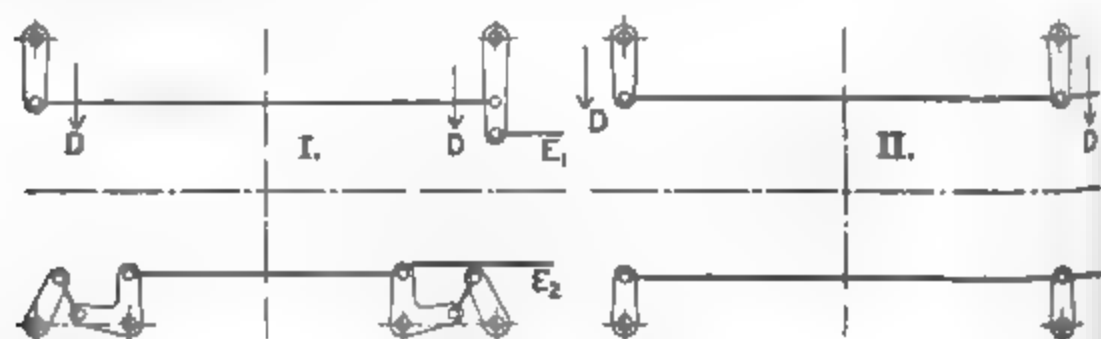


FIG. 475.—Gears without the Wrist-plate: I. Brown-Corliss; II. Laid and Bodley.

(k) THE USE OF TWO ECCENTRICS.—We have already stated in (e), that the valve must be released before the eccentric gets to its dead-point, and have shown in (g) that an eccentric to drive

ust-valves must have some advance, usually 15 to 20 degrees. condition is represented in Fig. 476 I., where  $\delta$  has the minimum e  $105^\circ$ , and it is assumed that latest dependable release must e place  $15^\circ$  before dead-center of the eccentric, or when the k is at  $60^\circ$ : this corresponds very closely with curve 4 on . 464.

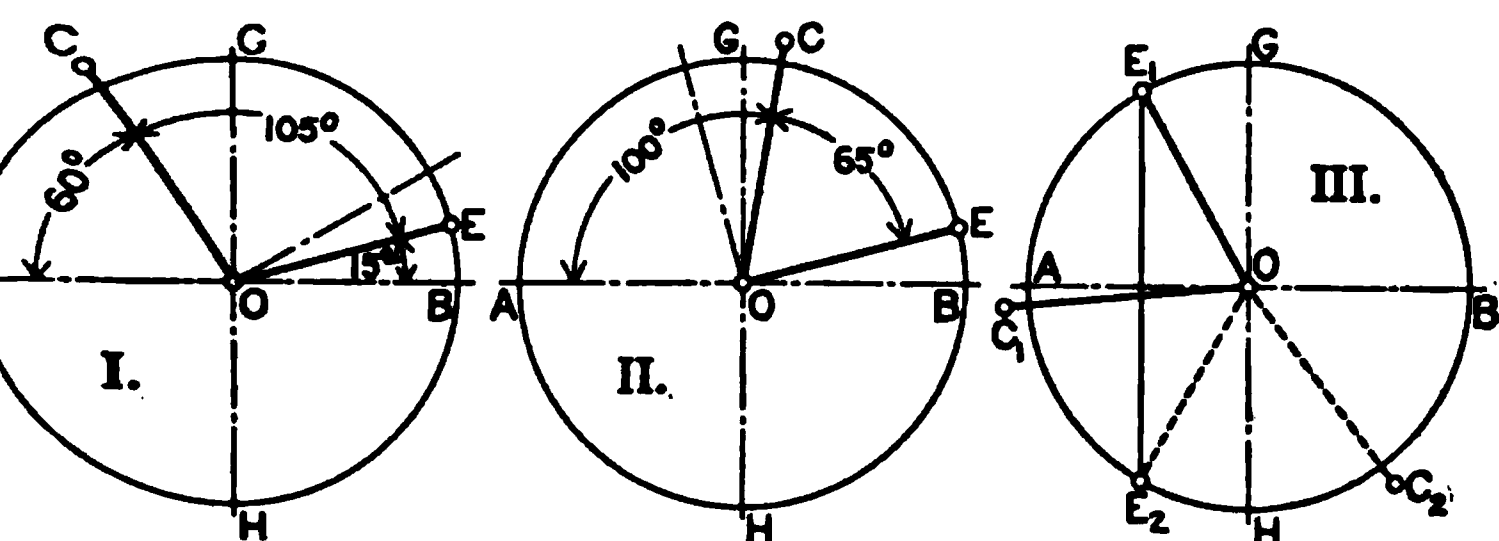


FIG. 476.—Diagrams of Eccentric-setting.

A very obvious way to increase the range of cut-off is to set the eccentric back, toward the crank, so that it will turn through a larger angle from the position for crank on dead-center to that for latest release. In Fig. 476 II. the negative advance or the angle of lag is  $25^\circ$ , and the latest cut-off begins with the crank about at  $100^\circ$ . It is entirely evident that this eccentric can be used for the steam-valves only.

The limit to the setting back of the eccentric is found in the fact that if the negative lap is made too great, the valve will not cover the port effectively when closed. Thus in Fig. 476 III.,  $C_1OE_1$  is the crank-eccentric from II. in the position for beginning of admission, and we see through what a small distance the valve has moved from its rest-position—a distance which is all the smaller if there has been much wrist-plate effect. This shows that the direct drive in Fig. 475 is highly appropriate if the eccentric is to be set well back of the 90-degree position. Fig. 476 III. brings out further the absolute necessity of insuring that a gear of this type shall never fail to release: for the condition which produced the indicator diagrams in Fig. 466 III. would in this case keep the valve open until the crank got around to  $C_2$ , or far into the exhaust-period.



(*l*) GEARS WITH MOVING TRIP-CAMS.—An alternate to the scheme of using a separate eccentric for the steam-valves is found in the rather recently applied idea of giving the knock-off

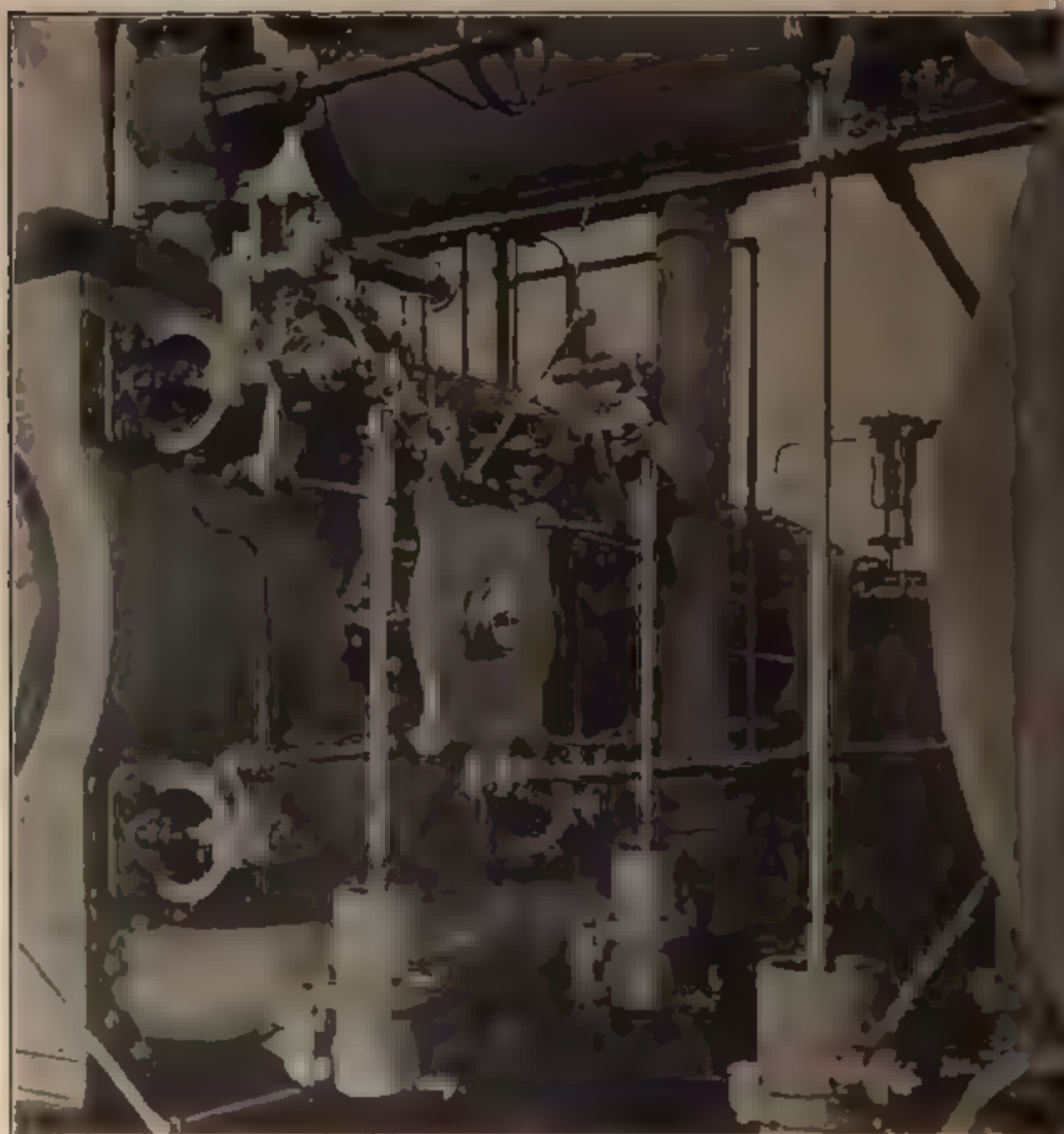


FIG. 477.—Extended Cut-off Gear, from the Snow Pumping-engine, Fig. 229.

cams an oscillating movement, so that they can catch and move the trip-arm even after the valve has started to return from its position of extreme opening. This type of gear is used chiefly on pumping-engines that deliver directly into the mains, working



must a moderate water-pressure ordinarily, but required to produce a much higher pressure when water is needed for fighting

A good example is illustrated in Fig. 477, and is further defined and discussed in Fig. 478.

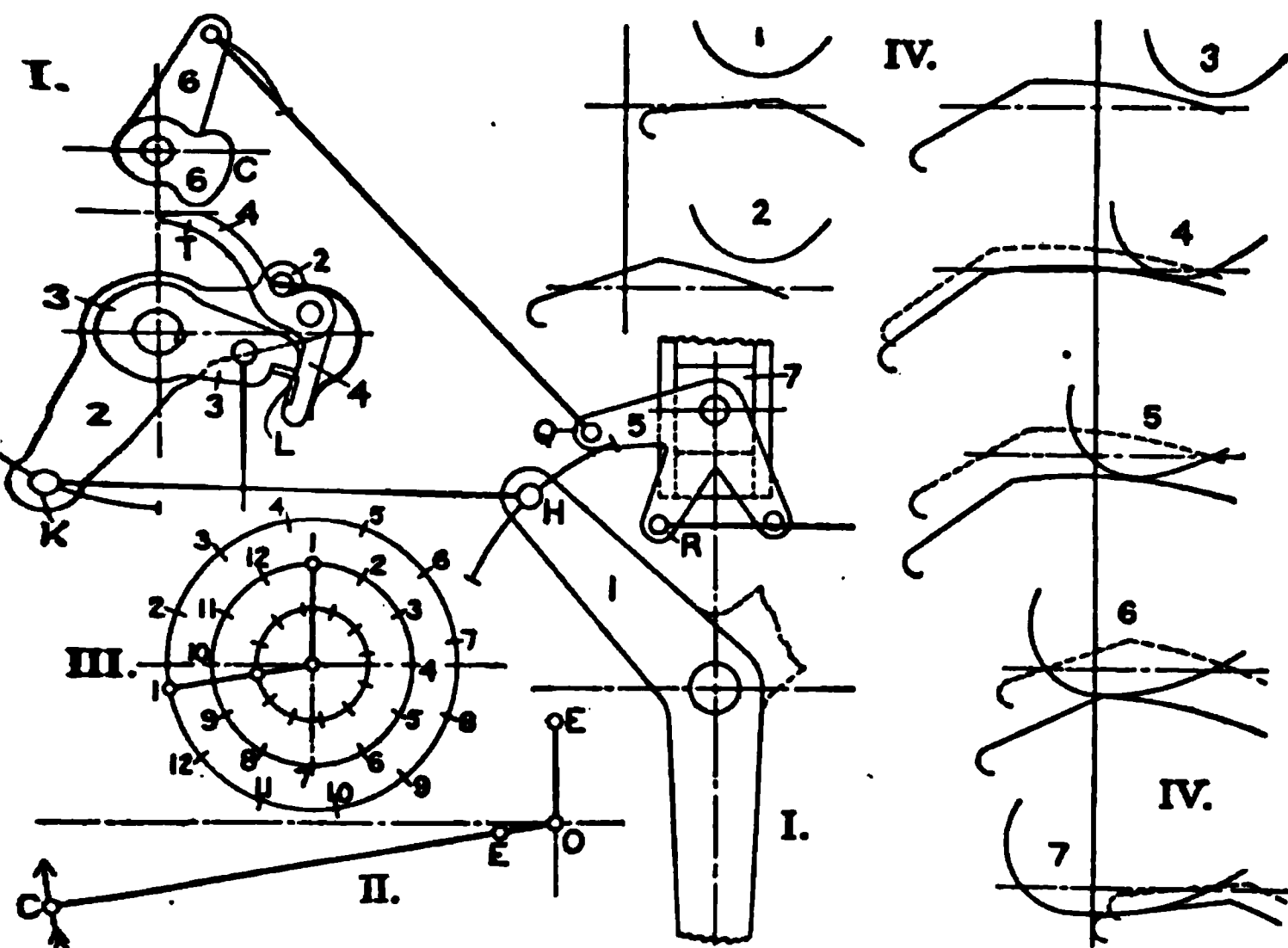


FIG. 478.—Action of the Snow Gear.

In Fig. 478 I. the gear for the head end is shown, and from the wrist-plate 1 to the hook-piece 4 it is of the usual form, except for the special shape of the tail T. The auxiliary rocker or wrist-plate 5 is driven from the main cross-head, the point R moving as if directly connected to the eccentric-center marked F on II. The block 7 is held and moved by the regulator, being raised to retard cut-off, lowered to hasten it. To illustrate the action of the oscillating cam C on piece 6, a series of successive positions of C and T is drawn at IV., the numbers corresponding to the angular positions of the crank-eccentric indicated at III. In cases 4 to 7, the dotted outline shows where T would be if not depressed by the cam C; and it is evident that release of the valve will take place between positions 4 and 5, when the crank

is near  $90^\circ$ . Note that T has its maximum travel to the left at 4, while C does not get to its lowest position till a little after 7.

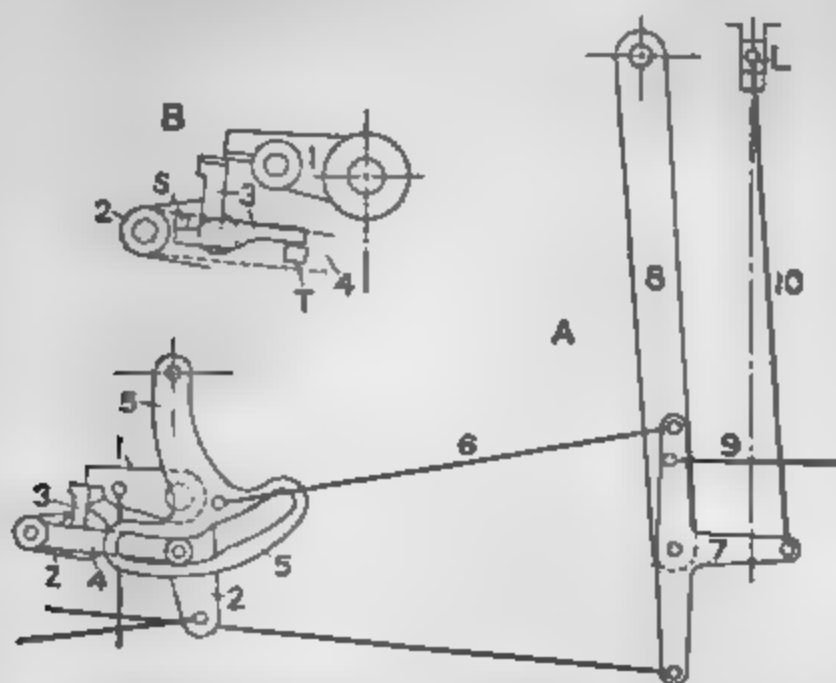


FIG. 479.—Nordberg Gear with Moving Trip-cams.

A device on this principle for an ordinary power engine is sketched in Fig. 479. The cam-piece 5 is pivoted at the top and moved by the oscillating lever 8, which is driven from a special eccentric on or near the crank-line, through the rod 9. The governor exerts its control through the T crank 7, the letter L marking the same piece here as on Fig. 461. The latch 3 cannot be fastened rigidly to the arm 4, since it must be free to spring back and let the end of 5 slide up past it. At B is shown how a horizontal arm fast to 3 can rest against the stop 8 on piece 2 ordinarily, and be raised at the other end by the projection T on piece 4 when the free end of 4 is raised by the cam-slot in 5. By allowing a slight clearance at this point, 4 is left free for the small movement due to the fact that the plain part of the cam-slot is not an arc central at the point about which the main arm 2 oscillates.

An interesting and unique valve-gear is shown in Fig. 480. The method of releasing is essentially the same as in Fig. 478; but the valve, instead of being closed by a dash-pot, is turned back positively by the oscillating cam-piece. The primary mech-

n, consisting of the free crank 2 and the disk 3 keyed to the e-spindle, is obvious. Here again the rocker 5 is driven from cross-head, with the effect of a center at F in II. The action of the cut-off gear is best seen from III., with the help of the detail

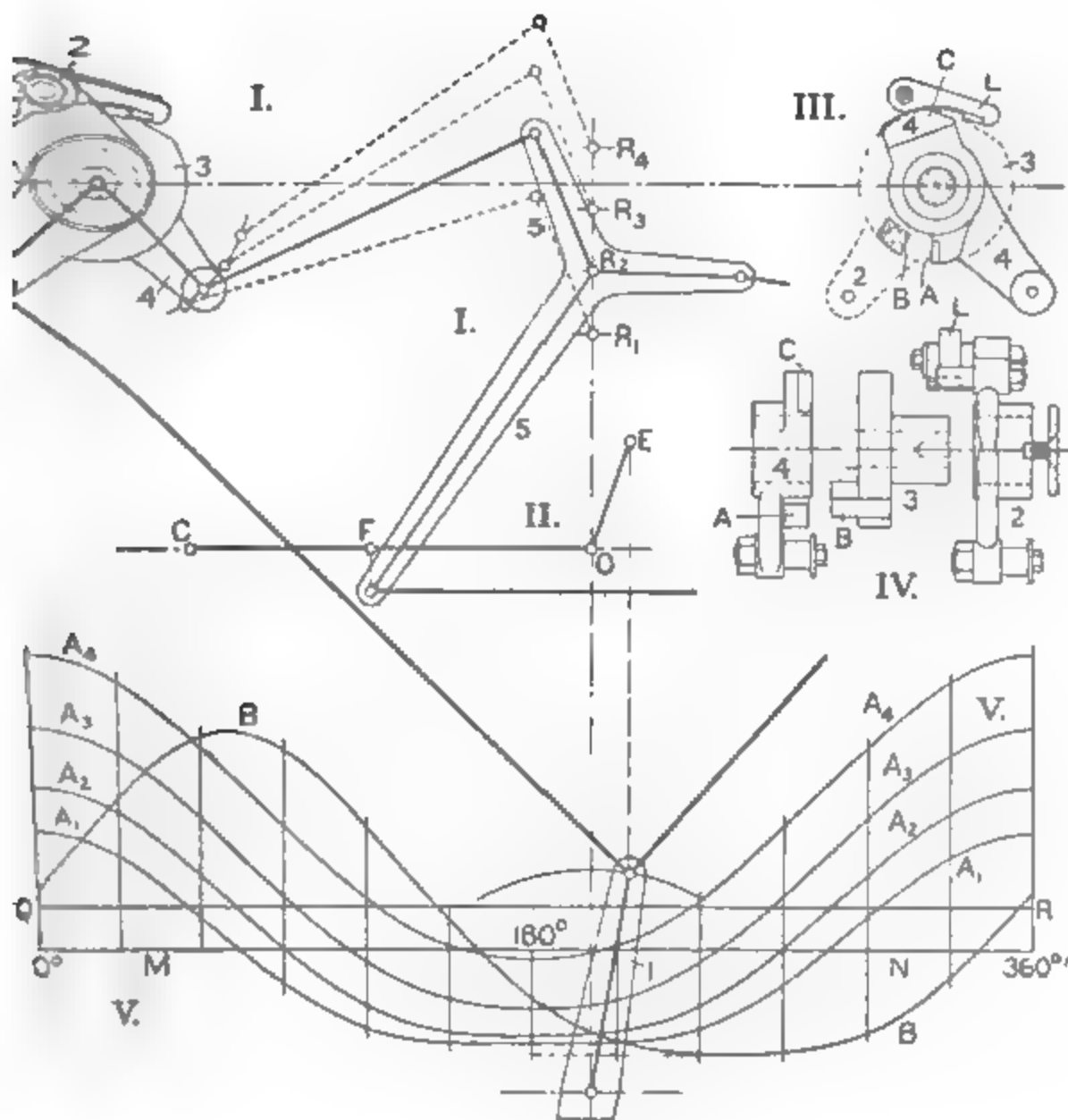


FIG. 480.—Steam-valve Gear of the Holly Pumping-engine, Figs. 227, 269.

sketches at IV. Just after the cam C raises the latch L clear of the hook-plate on 3, the arm A on 4 comes into contact with the stud B which projects back from 3, and pushes the valve back. To show the action fully, a set of motion curves is plotted at V. If pieces 2 and 3 were locked together with the latch engaged

as in I., the edge B would move according to curve B; the origin of the curve, laid off from MN as base, showing the curve distance of B from a mid-position determined by putting the eccentric at  $90^\circ$ . Similar curves for edge A, which also determines the movement of B, are given at  $A_1$  to  $A_4$ , showing travel from the same reference position that was used for B. It happens, the disk 3 has the same diameter as the valve, so these curves show valve-face movement directly. The lap-line is drawn at QR, and we see how the cut-off, shown by the intersection of the A curves with QR, is varied as the position of the regulator-block ranges from  $R_1$  to  $R_4$  in I. The critical point in the action of this valve-gear is on the side of closure. With earliest cut-off, curve  $A_1$ , the disk 3 must not be turned so fast that the hook cannot engage; while curve  $A_4$ , for latest cut-off, must dip below the lap-line QR far enough to secure effective sealing of the port. A disadvantage is that with late cut-off the hook is moving quite rapidly when it engages the disk, which involves a severe jerk on the machinery; but this is not of much importance, because the engine is required to develop its highest power only occasionally and for short periods, and a condition near  $A_1$  prevails normally.

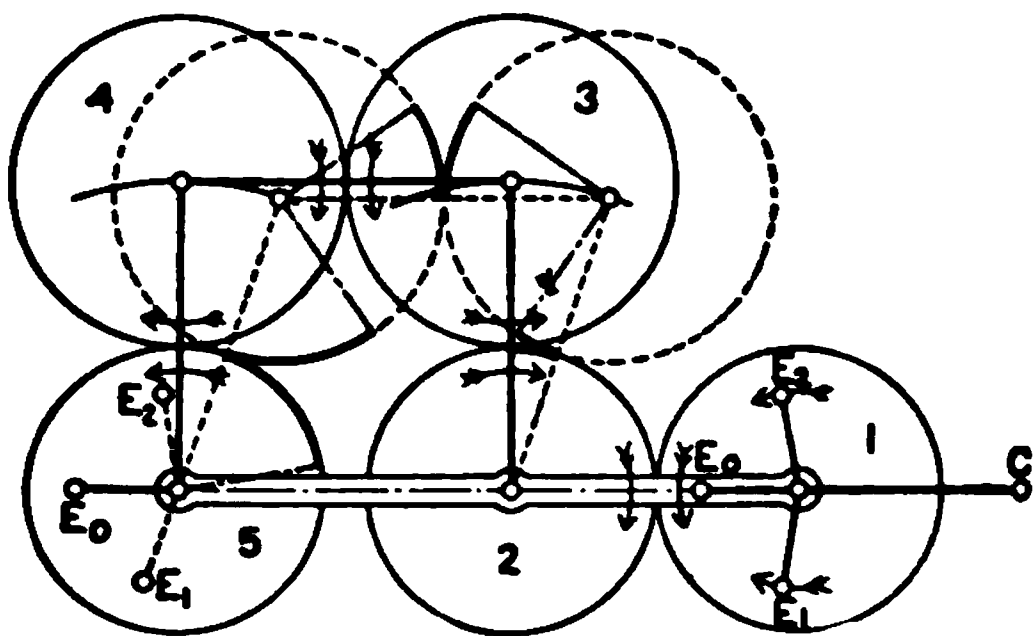


FIG. 481.—Scheme for Reversing the Corliss Engine.

m) A REVERSIBLE CORLISS ENGINE.—The gear outlined in Fig. 481 is taken, with some modification in the driving arrangement, from the description of a large mine-hoist in *Power for*

v., 1904. A link-motion could not be used in this case, because shortening the eccentric-radius will make the releasing-gear fail to act: instead, the eccentric-radius at full length must be rotated on the shaft from one running-position to the other, as indicated by gear-wheel No. 1 in Fig. 481, which wheel is either on the shaft or is made to turn with it. In the train 2 3 4 5, the intermediate bars 3 and 4 are carried at the corners of a jointed parallelogram; in the "mid-gear" setting of the eccentric  $E_0$  in 5, with reference to the crank on 1, the wheels are in the full-line position. By holding 1 and 2 fixed and swinging the frame to the right, as shown in dotted lines, the eccentric on 5 is moved to  $E_2$ ; and an opposite swing will carry it to  $E_1$ . To make clear the action of the mechanism, the radii of 3, 4, and 5 which originally lay under the bars of the frame are drawn in dot-and-dash on the second position; and the length of pitch-circle which has rolled away from each contact-point is indicated by heavy black arcs. It must be understood that this rigging is used only for complete reversal, the cut-off being determined by the usual trip-cams, under hand-control.

(n) DETAIL OF THE RELEASING-GEAR.—A number of typical devices are grouped together in Fig. 482. The same reference-numbers are used on all the figures, 1 designating the oscillating crank (skeletonized on all but VI.), 2 the latch-piece, 3 the valve-arm, and 4 the cam-piece: the parts are lettered mostly as on Fig. 460,  $C_1$  marking the trip-cam and  $C_2$  the safety-cam. The rod driving 1 is marked W when it goes to a wrist-plate, E when the direct drive of Fig. 475 is used. Design I. is an improvement on the standard rig in Fig. 460, in the direction of quiet and easy running; the tail of the hook carries a roller, and this works on non-metallic surfaces, the cam-blocks being made of the hard leather-like composition known as "fibre", and the same substance used to face the cam-ring between the blocks and for a little pad under the stop-pin S. At II. we see another case of the use of the roller, the trip-cam also being of this form. In III., typical of several designs, the trip-arm takes the form of a long tail, the side of which slides on the cams. IV. shows a case of hooking fast from the inside, while in V. the cam is of the positive-movement

type, so that the latch-block must be free to slide up and down. Finally, VI. shows a design with the fixed catch-plate on piece 1 and the latch-blocks in 3; this latch is a simple rectangular piece of steel, fitting easily in the slot in 3, dropping into place by gravity.

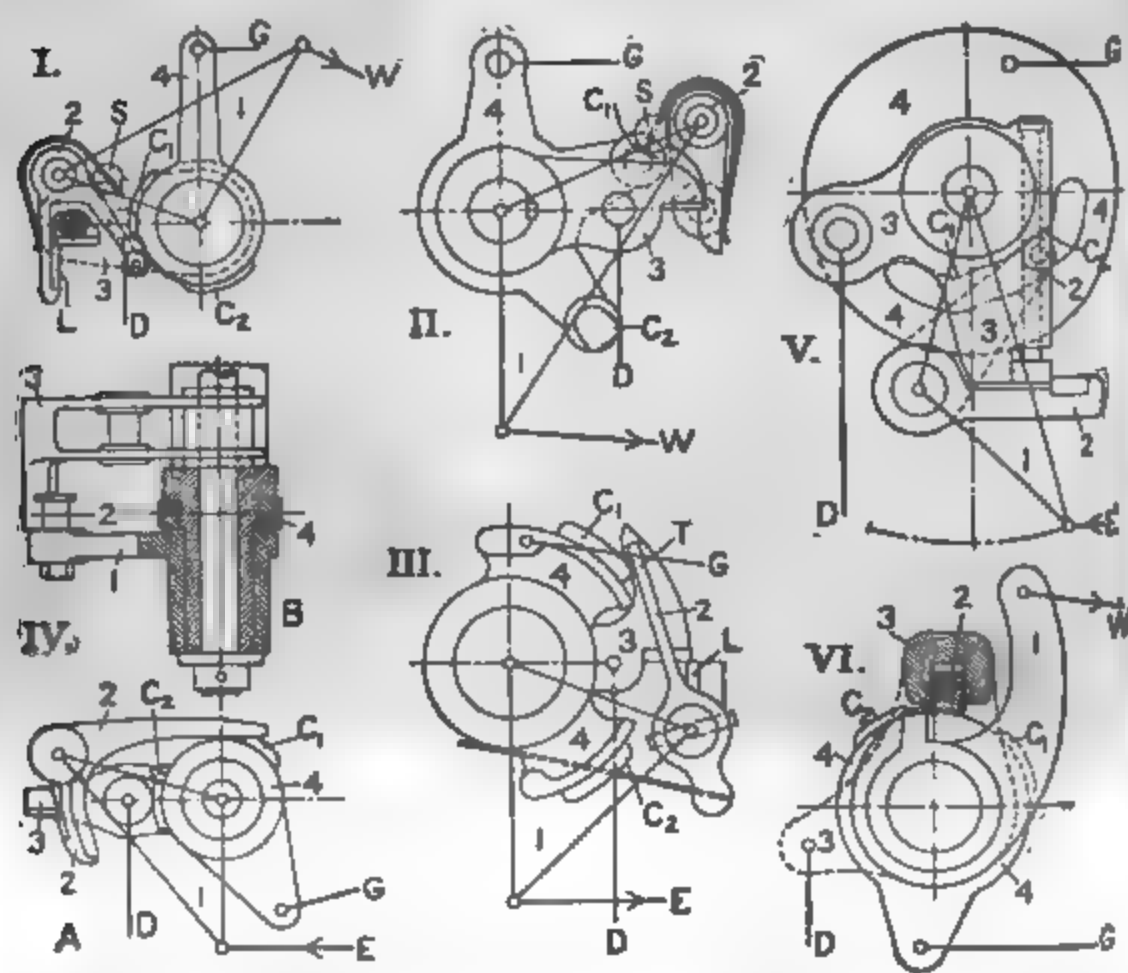


FIG. 482.—Various Releasing-gears. I. Murray-Corliss; II. Fishkill; III. Brown; IV. Lane & Bodley; V. Harris; VI. Robinson.

alone, and supported by a roller at each end, these resting on double cam-rings.

*Inside and Outside Valve-arms.*—An important distinction in the arrangement of the cut-off gear is brought out by comparing IV. B in Fig. 482 with Fig. 460. In the latter, and in all the designs here shown except III. and IV., the valve-arm is at the outer end of the spindle, beyond the bracket. The principal forces can be brought closer to the cylinder (and hence be less severe upon the valve-bracket) by putting the keyed valve-arm inside of all the rest of the gear, in the opening of the bracket

which gives access to the stuffing-box—this opening being suitably enlarged. In every case, however, the oscillating crank and the driving-gear are carried on a stud projecting from the bracket.

Some European designers have put the releasing-gear in the driving-rods (marked DC on Fig. 462) instead of between two oscillating cranks, changing the latch-mechanism as necessary; but this plan seems to have little to recommend it.

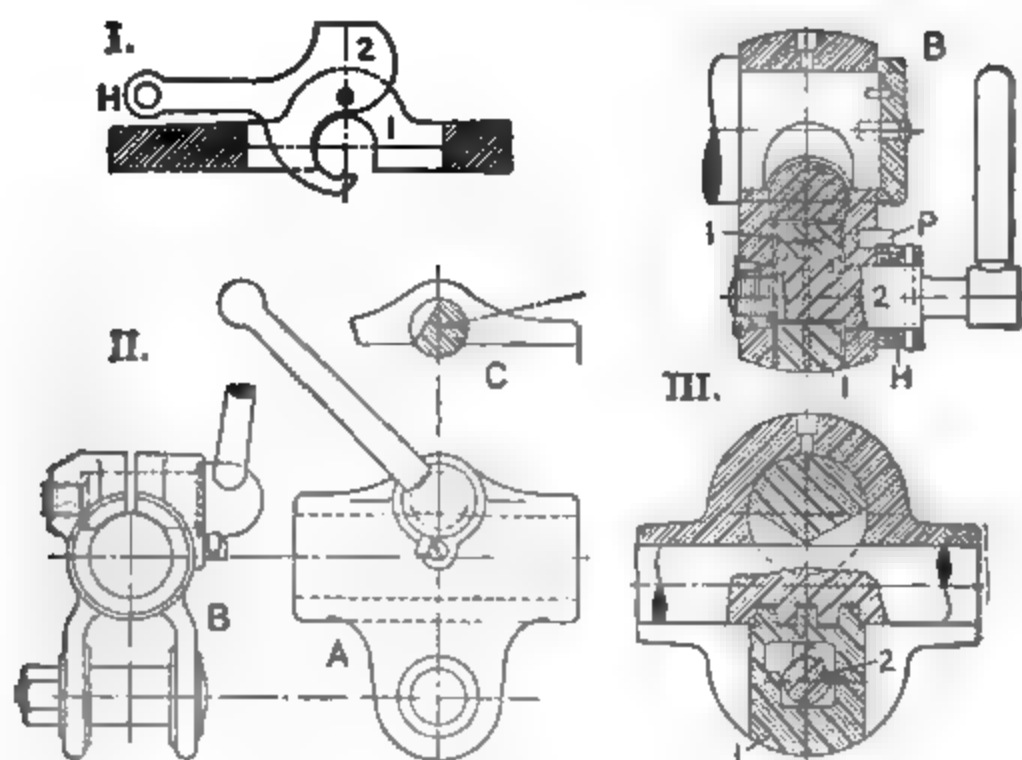


FIG. 483.—Different Unhooking Devices: I. Murray; II. Harris; III. Brown.

(c) DEVICES FOR RELEASING THE DRIVING-ROD.—An improvement on Fig. 458 C is shown in Fig. 483 I., the cam-plate holding the rod down when in the running position and lifting it off the pin as the handle H is swung to the right. Devices which take an even firmer hold, with less tendency to wear loose and rattle, are given at II. and III. The first belongs to a gear arranged, at the cylinder, like those in Fig. 475, the block being at the middle of the horizontal rod connecting the two oscillating cranks. In both, the first movement of the handle, from the free position, engages a positive catch; and further movement clamps the split block tightly upon the rod. In III., the cam 2 is held down

(joint open) by friction of a slight elevation on the hub H, under the pin P: raising the handle a little frees the hub, whereupon the spring inside of H throws the block 1 up as soon as the teeth come into the engaging position: and after this automatic engagement the block is clamped by hand. Another scheme is to make the wrist-plate of two disks, one driven by the reach-rod, with some arrangement for locking the disks together.

(p) THE DASH-POT.—The two typical forms of this device are shown in Figs. 484 and 485, with single and double plunger respectively.

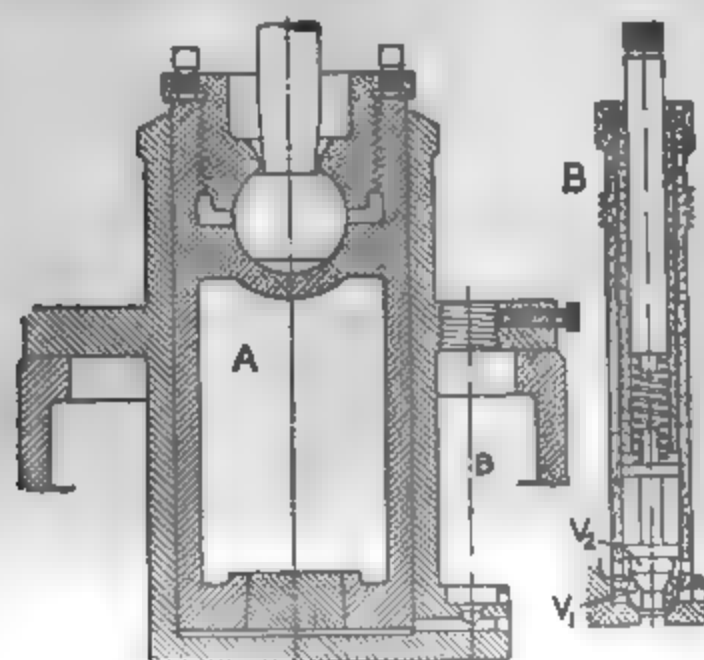


FIG. 484. — A Single-plunger Dash-pot, Frick Design.

The dash-pot has two functions: the first, to develop a strong force to pull the valve shut quickly; the second, to bring the moving parts quietly to rest after the valve is closed. The name comes properly from this second function, chiefly because the earlier engines had weights or springs to close the valves, and the dash-pot was then only what its name implies.

A possible arrangement consists of a simple plunger, with a small cock to regulate the necessary flow of air. To get a wider and more flexible adjustment, several valves are used, and in the majority of cases a second plunger is added, as  $P_2$  in Fig. 485 which takes most of the cushioning function, leaving the vacuum action chiefly to  $P_1$ .

As the plunger rises, a partial vacuum is formed beneath it, and some air is drawn in; as it nears the bottom of the cylinder, on the drop, this air is compressed and furnishes the needed counterforce. In both figures we see first an adjustable valve  $V_1$ , next a check-valve  $V_2$ , so that the egress is made freer than the ingress, to allow for the fact that the downward movement is quicker



in the rise. During the long rest between admissions the excess of air can all escape, until only atmospheric pressure is left in the small clearance-space beneath the plunger. The admission-valve  $V_1$  on Fig. 485 does not close its passage even when

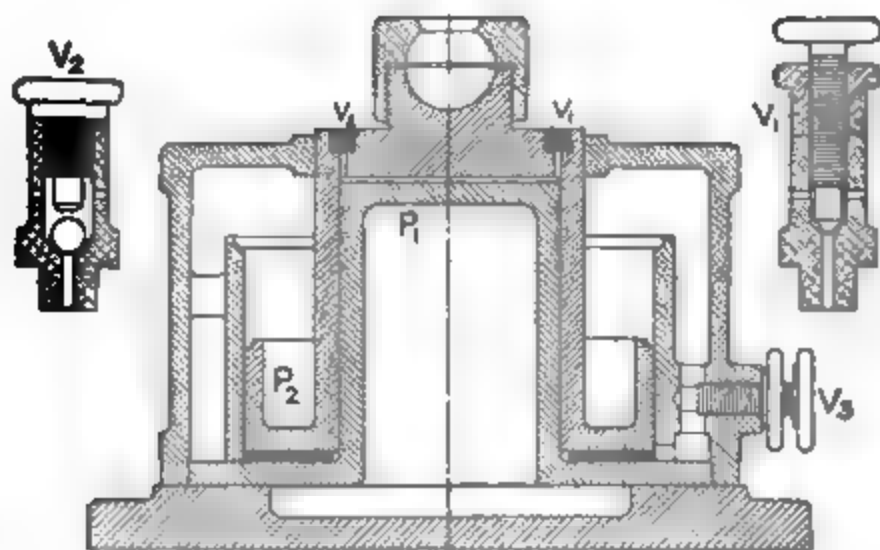


FIG. 485.—A Two-plunger Dash-pot, Vilter Design.

screwed all the way in. Relative to the piston-displacement, this valve is larger than  $V_1$  and  $V_2$ , so as to permit a freer inflow under the outer plunger, which therefore exerts less pull on account of vacuum, and is more strongly checked on the down stroke. It must be understood that these devices are based much more on experience and trial than on *a priori* reasoning—both as to form and as to proportions.

### § 57. Releasing-gears with Gridiron Valves.

There are a number of standard engines which have flat grid-iron valves moved by releasing valve-gears. In the general arrangement of the valves and in the form of the mechanism they show greater variations than do the more numerous Corliss designs; but they are sufficiently represented by the two examples now to be described.

(a) THE GREENE VALVE-GEAR.—In Fig. 486, view A shows the special gear for the steam-valves, the slide S being carried on a bracket from the cylinder and moved by the rod E. The catches

C. C, adjusted as to height by the governor-gear G, engage immediately the trip-toes T, T, and push the valves open: the latter closed, after release, by the steam-pressure on the ends of the large valve-rods and by the drop-rods D, D. By making

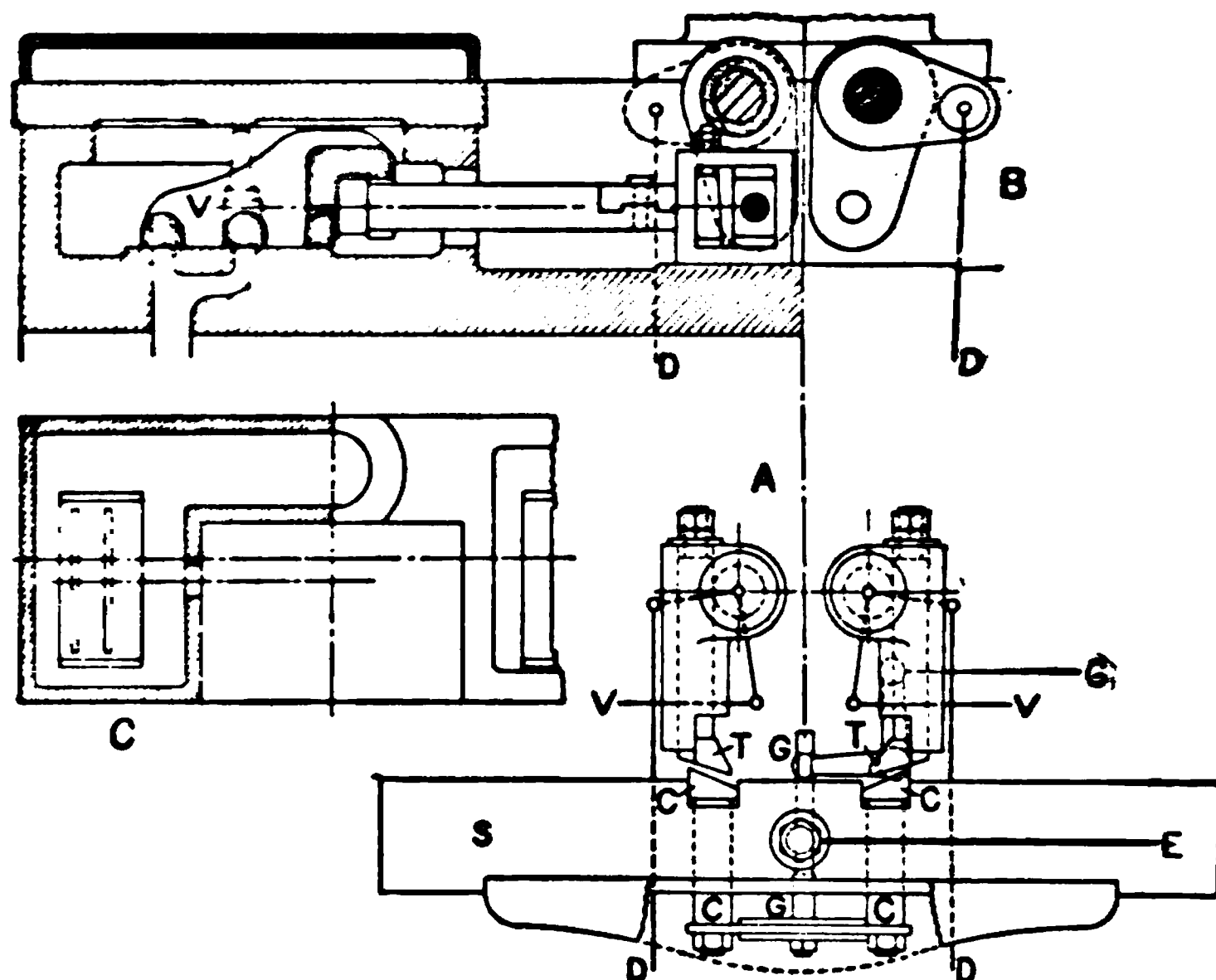


FIG. 486.—Valve-gear of 20'' x 42'' Greene Engine. Scale 1 to 12 and 1 to 30.

toes T, T, of slightly unequal length, the cut-offs can be equalized against the distortion in piston-movement caused by the connecting-rod. View B is a partial section of the cylinder and valve, showing the push-cranks and the drop-rod arms on the oscillating shafts which transfer movement from the catch-gear to the top of the cylinder. Note the adjustable bearing for the slide-block, in the valve-rod end. View C is partly a top view, partly a section above the valve-seat, to show how the steam- and valve-chambers are formed upon the cylinder, in such a way as to leave a clear space for the valve-gear. The exhaust-valves are set crosswise as in Fig. 264, and driven, with a strong distortion from harmonic

motion, by an oscillating shaft which lies parallel to the axis of the cylinder.



FIG. 487. -Valve-plug and Gear, Wheelock Engine, see Figs. 214, 266.

(b) THE WHEELOCK VALVE-GEAR. The valve-plug belonging to the cylinder in Fig. 266, with all the gear that is mounted upon it, is shown in Fig. 487, while the detail of the mechanism is given in Fig. 488. The valve-pusher 2 carries the hook-block H, which engages the latch L on the slide 3, the latter being free to move up and down between shoulders on the head 4 fast to the valve-rod. As better shown in view B, the latch L has projecting ends which rest upon the lifter 5, and this is raised and lowered by the cam 6. This piece is swung one way or the other by the governor; the part marked  $C_1$  regulates the cut-off, while  $C_2$  is the safety-cam, acting just as in a Corliss engine, and raising the latch L so far that H cannot engage it. Latest cut-off is indicated by dotted-line positions of H and L and of the arm 2 on view A, the greatest valve-travel, with the dimensions shown at C, being  $1\frac{1}{8}$ ". Note how small is the lap of this valve, on both sides of the port, when closed.

The steam-valve is closed by steam-pressure on the end of the valve-rod and by the spring, with perhaps a little help from

the dash-pot: but the latter serves chiefly as a cushion, with a leather pad to limit the stroke of the plunger. The toggle-joint mechanism for the exhaust-valve is merely outlined on this figure: note that the rod for this valve has a good bearing in the head of the bracket, alongside of the dash-pot. In B the sectional outline of this head is dotted upon the full view taken farther in.

The action of these gears differs in no essential from that of the Corliss gear: but one point to be remarked in both is that the

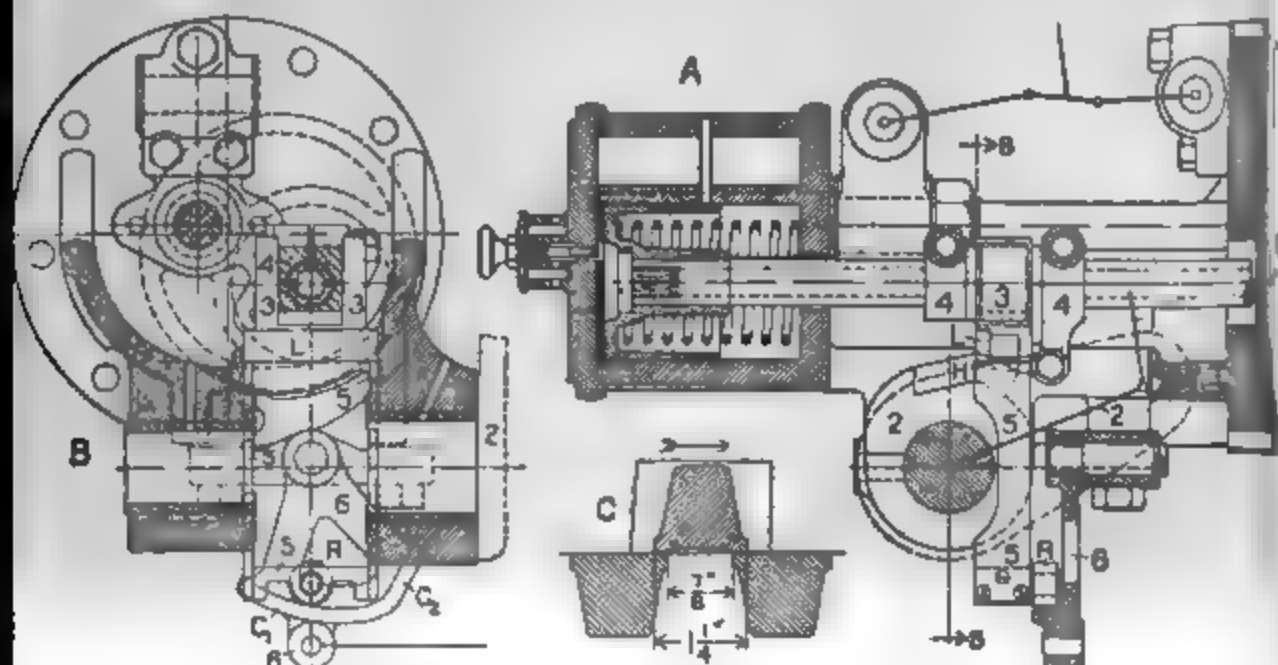


FIG. 488.—Details of Wheelock Gear. Scale 1 to 8.

piece carrying the catch-plate has harmonic motion, as is likewise the case in Figs. 475 and 482 V. We have here illustrated two types of gridiron valve, the first in Fig. 486, with a few long ports and a sidewise movement; the second valve is long and narrow with many short bars set across and with the movement lengthwise. The latter type is more often used; and in several designs admission-valves of this form are placed vertically on the side of the (horizontal) cylinder, with the exhaust-valves beneath the cylinder, as described under Fig. 486.

### § 58. Non-harmonic Gears with Variable Eccentric.

(a) THE PORTER-ALLEN VALVE-GEAR.—The primary mechanism in Fig. 489 I., consisting of the eccentric OE, the oscillating link

**EAD** or 1 and the carrying-rod 2, is a radial valve-gear. That the movement of D, or of any point B in the slot, is the resultant of two component eccentrics is better shown at II. The mid-position  $A_0D_0$  of the link is determined by bringing E to O; if now this center E on the rod AE were brought to B, the link would move to  $A_1D_1$ , every point on it receiving the horizontal displacement of the eccentric-center E. Swinging the rod to  $A_1E$  rotates the link to  $A_1D_2$ ; and it is easy to see that the additional travel  $D_1D_2$  is equal to the horizontal projection  $FE_1$ .

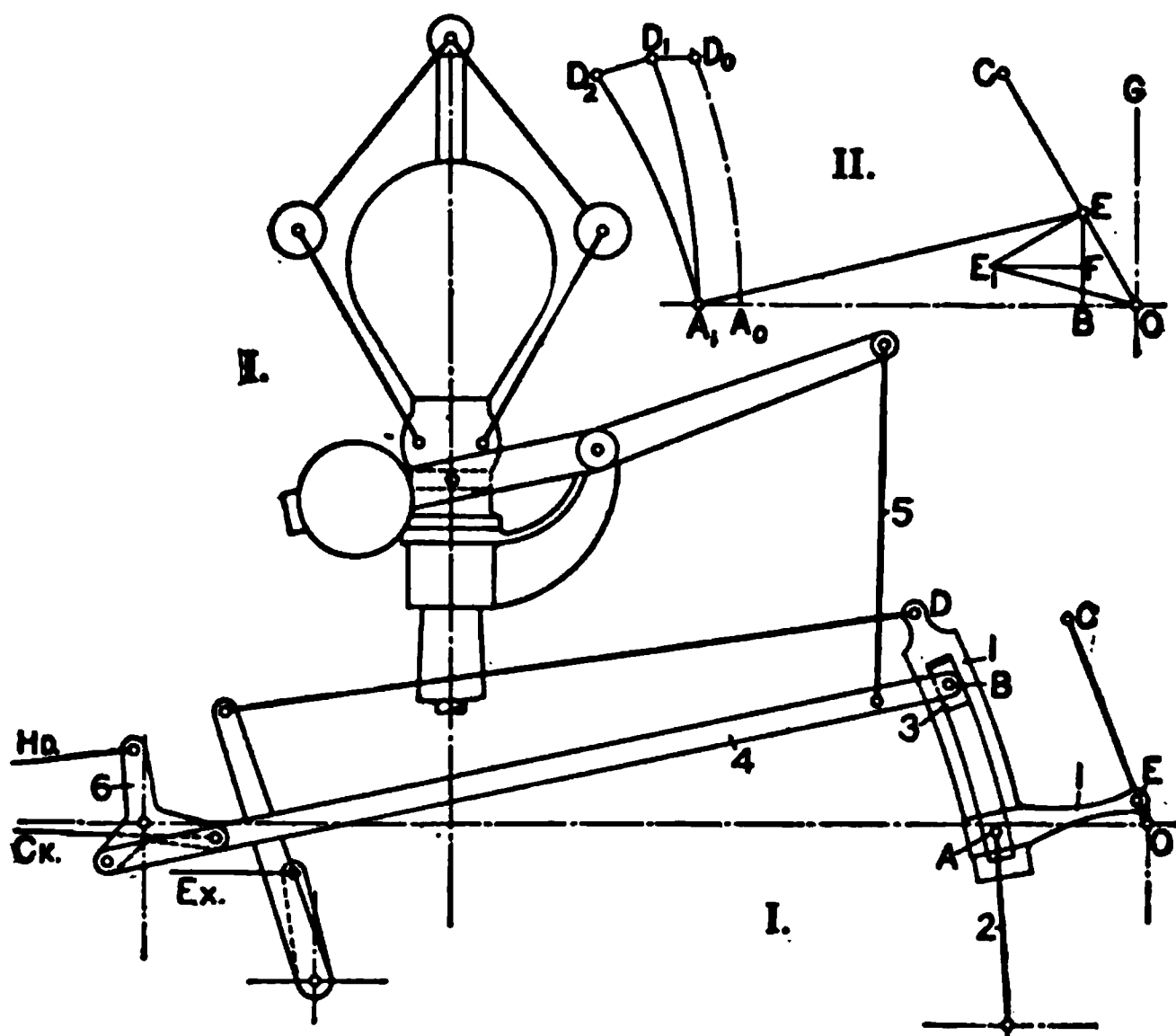


FIG. 489.—Outline of Porter-Allen Valve-gear.

of an eccentric  $EE_1$ , perpendicular to  $OE$  and in length equal to  $OE \times AD/AE$ . Then the resultant eccentric is  $OE_1$ ; and as the variable driving-point B moves along the arc AD, the effective eccentric-center will move along  $EE_1$ . The eccentric-rod AE is so short that its effect upon the horizontal movement of A must be considerable.

In the actual mechanism, the link AD is formed right upon one half of the eccentric-strap. The exhaust-valves, coupled on

one rod as in Fig. 250, are driven from D; while separate rods from the two arms of the wrist-plate 6 give the steam-valve a movement which is very decidedly shortened up on the down side. The action of the governor is obvious.

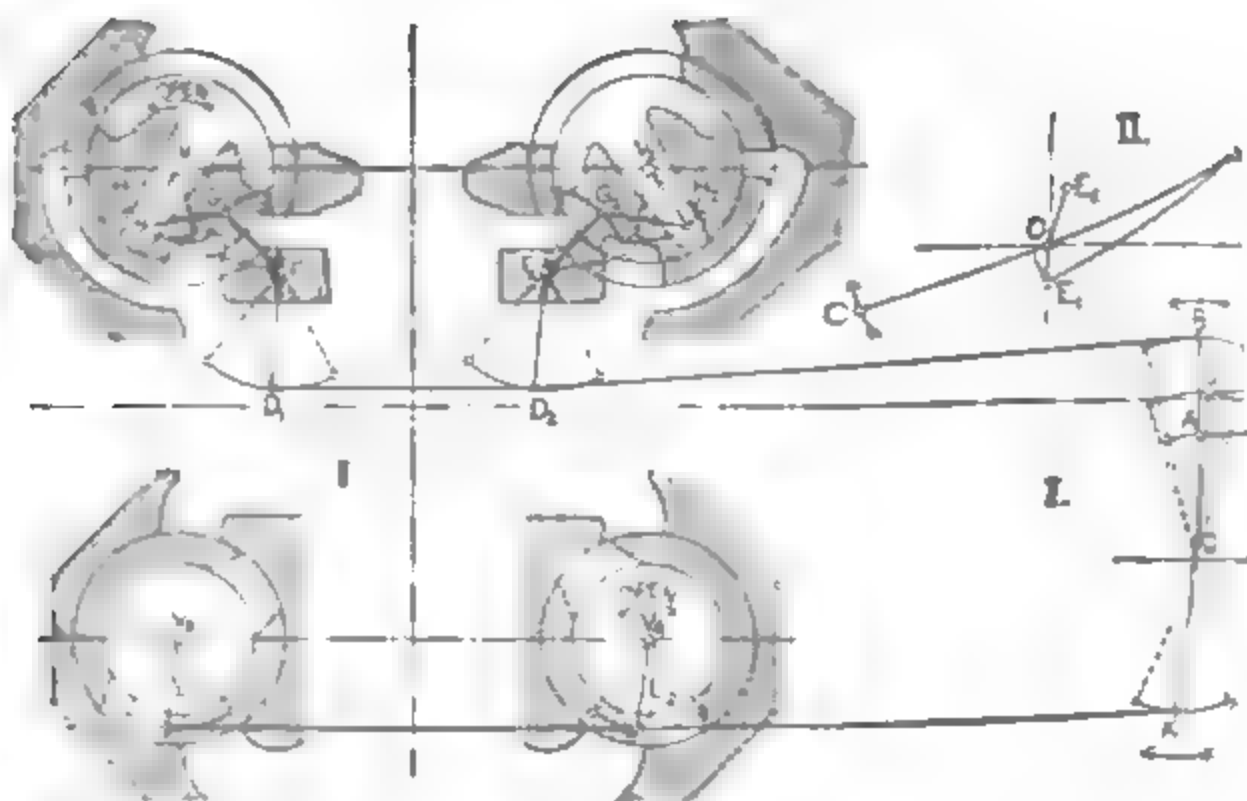


FIG. 490.—Outline of Fleming four-valve Gear, for 16''x16'' Engine.  
Scale 1 to 14, 1 to 7.

(b) THE FLEMING VALVE-GEAR, shown on Figs. 215 and 263, is drawn in outline for a simple engine in Fig. 490, the valves being made double-size. The bent levers DFG act as individual wrist-plates for the steam-valves, producing a strongly non-symmetrical movement, as is shown by dotting in the extreme positions. The eccentric-setting is indicated at II. The steam-valves are directly driven at H—refer to § 57 (j)—but the indirect wrist-plate connection, with a direct rocker-arm QAB, makes the eccentric indirect at E<sub>1</sub>. The exhaust-valves here have a movement opposite to that in the ordinary Corliss gear; but the reversing-rocker KQJ balances this, so that the fixed exhaust-eccentric OE<sub>2</sub> has the direct setting.

The first thing to do with a gear of this sort is to plot a general diagram for the movement of the mechanism at the cylinder,

without regard to the eccentric, as in Fig. 491. Here the abscissa, measured from EF along AB, is the horizontal displacement of the point A on the steam rocker-arm; the ordinate, measured from AB, is the corresponding travel of the valve-edge along the

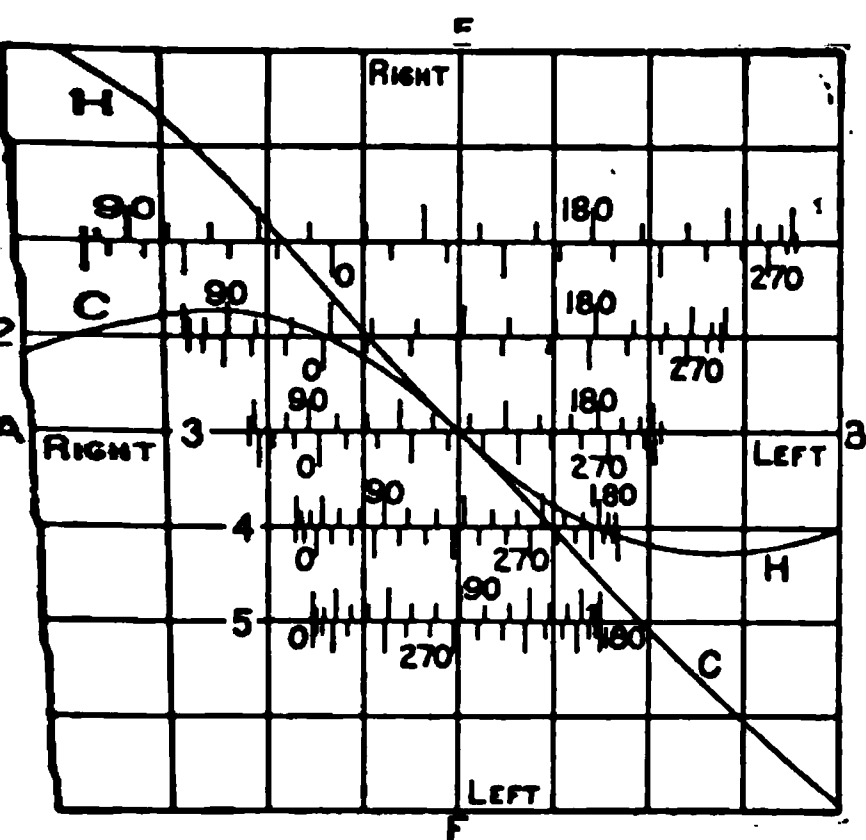


FIG. 491.—Motion Diagram for Gear on Cylinder

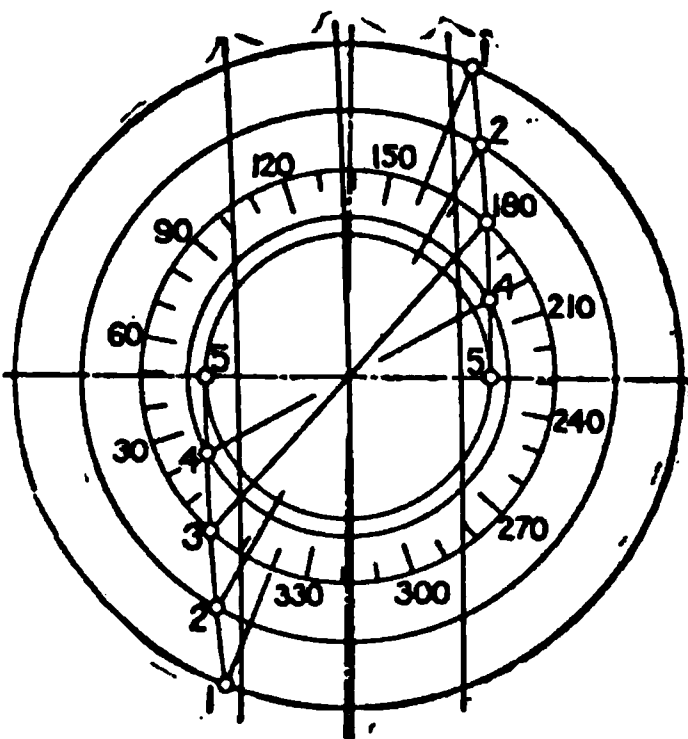


FIG. 492.—Eccentric Diagrams. Scale 1 to 2.

curved seat. The mid-position from which all travels are measured is drawn in full lines on Fig. 490, for the eccentric at 90°. The head-end curve H has travel for opening laid off upward, the crank-end curve C downward.

The next step is to draw a set of plain eccentric diagrams like Fig. 357 I. (compare Fig. 359 as to general arrangement). In Fig. 492 the E-locus is laid off with the known dimensions of the governor, and the whole range of movement is divided into four equal parts. On each eccentric-circle (as on no. 3 only in the figure) must be laid off an angle-scale, with the zero where the eccentric is when the crank is on head-end dead-center. The reference-line being an arc struck with the eccentric-rod as radius, the actual displacements of A can be very exactly measured off directly from this figure. Distances thus got are laid off on the base of Fig. 490, being indicated by the several displacement-scales there marked, which show positions of A corresponding to crank-positions designated by the angle-numbers.

Ordinates from Fig. 491 are now used to plot curves on the developed crank-circle, as in Fig. 463 (and Fig. 337). To avoid overlapping and confusion, separated base-lines are used in Fig. 493,  $A_1B_1$  for the head end,  $A_2B_2$  for the crank end. The effect of the difference in the angles  $D_1F_1G_1$ ,  $D_2F_2G_2$  on Fig. 490 is seen in the shorter travel of the crank-end valve. With this shorter travel can very well be used a smaller lap than in the head end, so as to get equalized cut-offs with practical equality in the effective widths of opening.

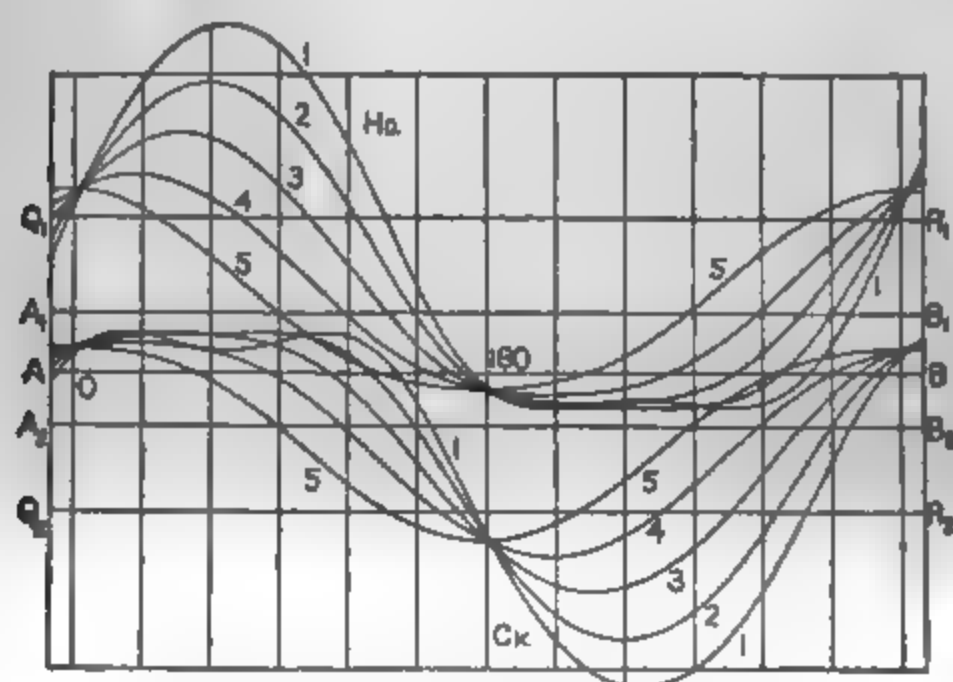


FIG. 493. Curves of Valve-movement.

The action of the triple-ported steam-valves in this engine has already been discussed, under Fig. 470. The exhaust-valves call for no additional comment here, having been illustrated in Fig. 472; some remarks under Fig. 475, as to the advantage of retaining the wrist-plate effect in the exhaust-valve drive, are equally applicable to this gear.

(c) THE BALL FOUR-VALVE GEAR.—The most highly developed design with four valves is partly shown in Fig. 494. By putting the valves in the cylinder-heads more freedom in arranging the ports is gained, so that it is easy to get triple-ported steam-valves with full effect of all three ports, and double-ported through-



passage exhaust-valves. The valve-gear on the cylinder is outlined at II.; and for the admission-valves the idea of a rest during closure with a quick movement for opening is very perfectly applied. The gear is of the individual type, with everything except the main driving-arm 1 inside of a tight casing filled with oil. In this mechanism there is first a "quadric chain" with pieces 1, 2, and 3 as the moving parts (laid out in plan at III.), the oscillating crank 3 turning freely on the valve-stem. The

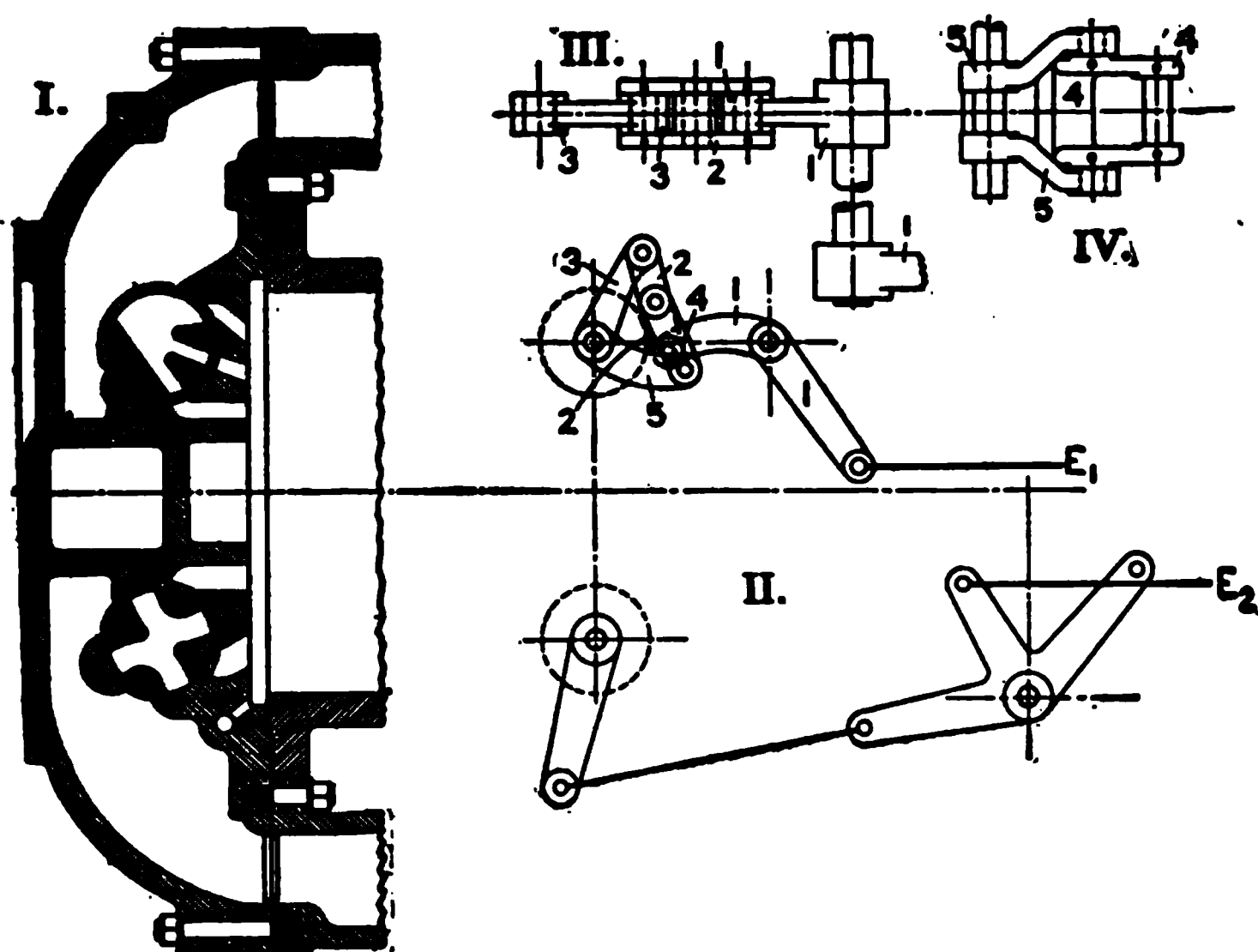


FIG. 494.—Outline of Ball Four-valve Gear.

valve is driven from the middle of link 2, through the pieces 4 and 5 laid out at IV., 5 being keyed to the valve-stem. Each of these pieces is a malleable casting, made up of two links (for the sake of symmetrical force-action) joined by a cross-bar placed where it will not interfere with the movement.

The action of this gear is brought out by Fig. 495. The driving-arm OF is vertical in mid-position, so that OA has a symmetrical oscillation. Equally-spaced positions of the mechanism are plotted, for a range of movement a little greater than the

widest actual movement, as indicated by drawing the largest and smallest (reduced) eccentric-circles on  $OE_1$ , and projecting to the path of A. The path of C from 1 to 6 coincides almost exactly with a circular arc struck from the highest position of D: and there is therefore, as shown by the curve at II., absolute rest of the valve during nearly half of the cycle-period—this

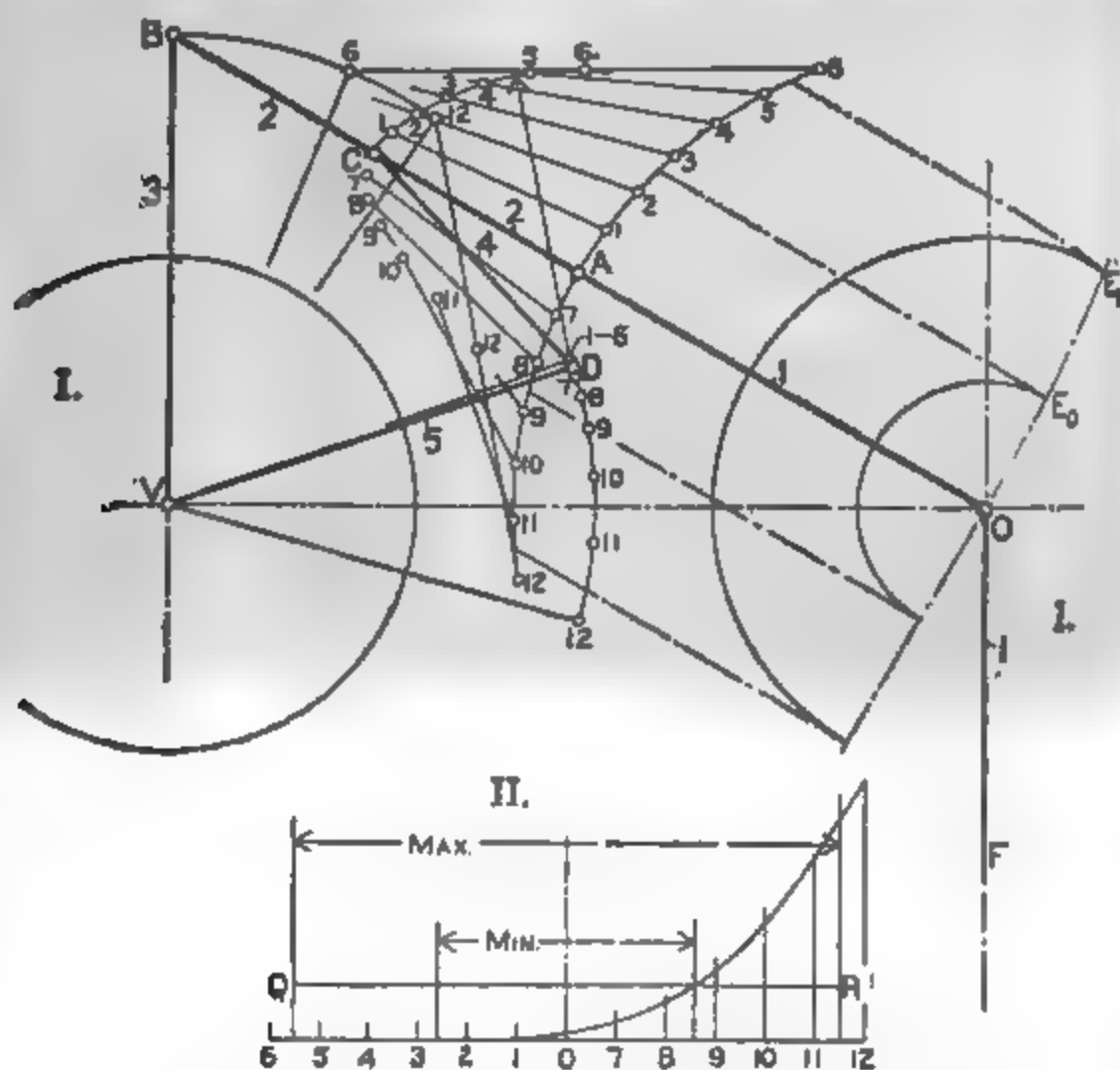


FIG. 495.—Action of Steam-valve Drive.

latter diagram being in the same terms as Fig. 491, with travel of A as base, travel of D as ordinate. For an unbalanced, single-function valve this action is ideal.

(d) THE MCINTOSH AND SEYMOUR VALVE-GEAR.—The valve-gear belonging to the engine in Fig. 218, of which some details can be seen on Figs. 264 and 265, is drawn in outline at Fig. 496 I. The main eccentric  $E_1$  acts through the bent rocker-arm ABD

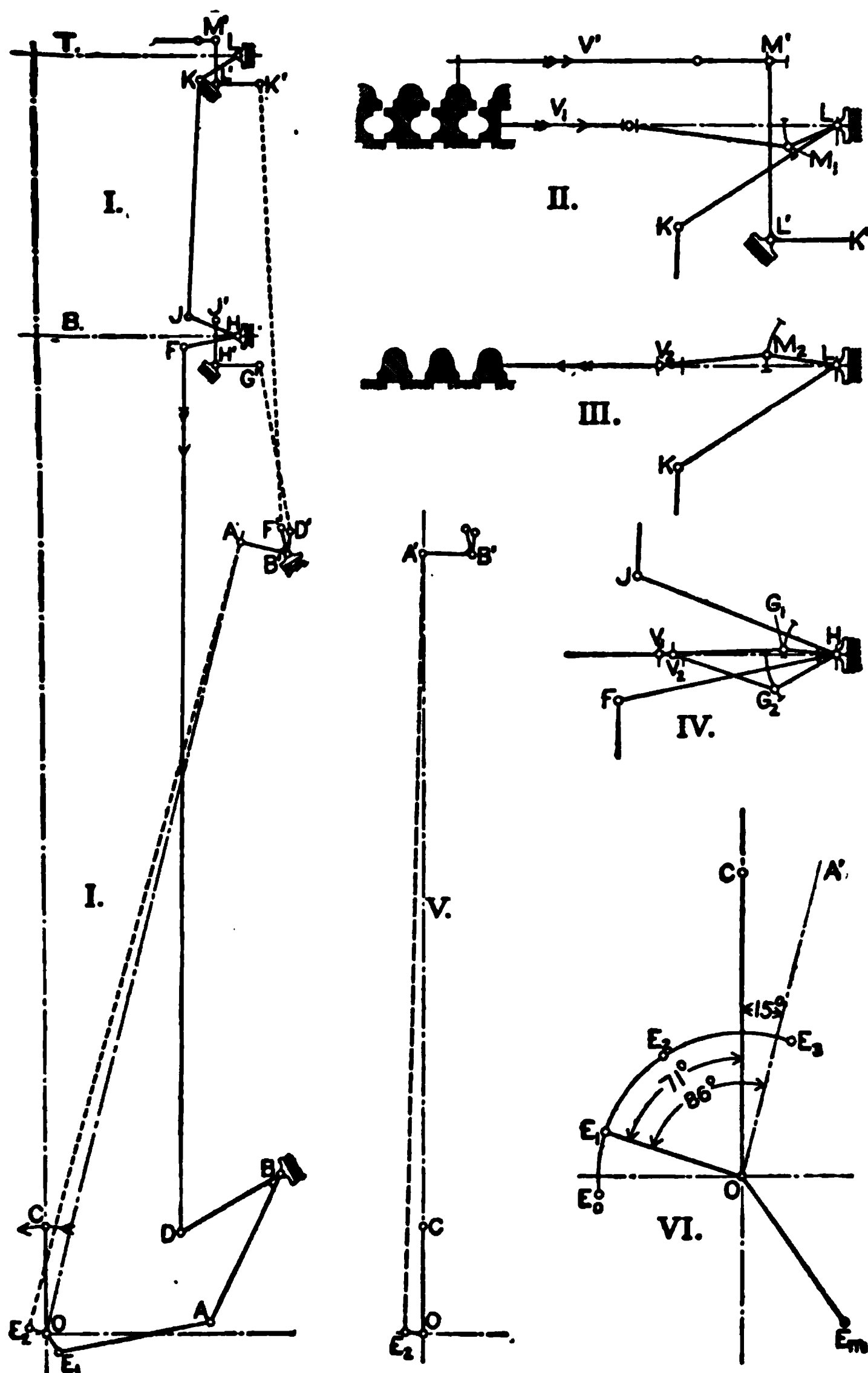


FIG. 496.—Outline of McIntosh and Seymour Six-valve Gear.

to drive the main steam-valves and the exhaust-valves, as shown at II. and III., which diagrams are both for the top or head end of the cylinder; while the cut-off eccentric  $E_2$ , adjusted by a shaft-governor as at VI., controls the riding cut-off valves. Diagram IV. is an outline of the main valve-drive at the bottom end,  $V_1$  again representing the steam-valve and  $V_2$  that for exhaust. At V. we see how, for convenience in graphical work, the cut-off eccentric and its rod and rocker  $A'B'$  can be brought to a vertical stroke-line without change in effect. The discussion of this mechanism will be chiefly analytical, but in some parts synthetical, with the description of the several diagrams made as concise as possible.

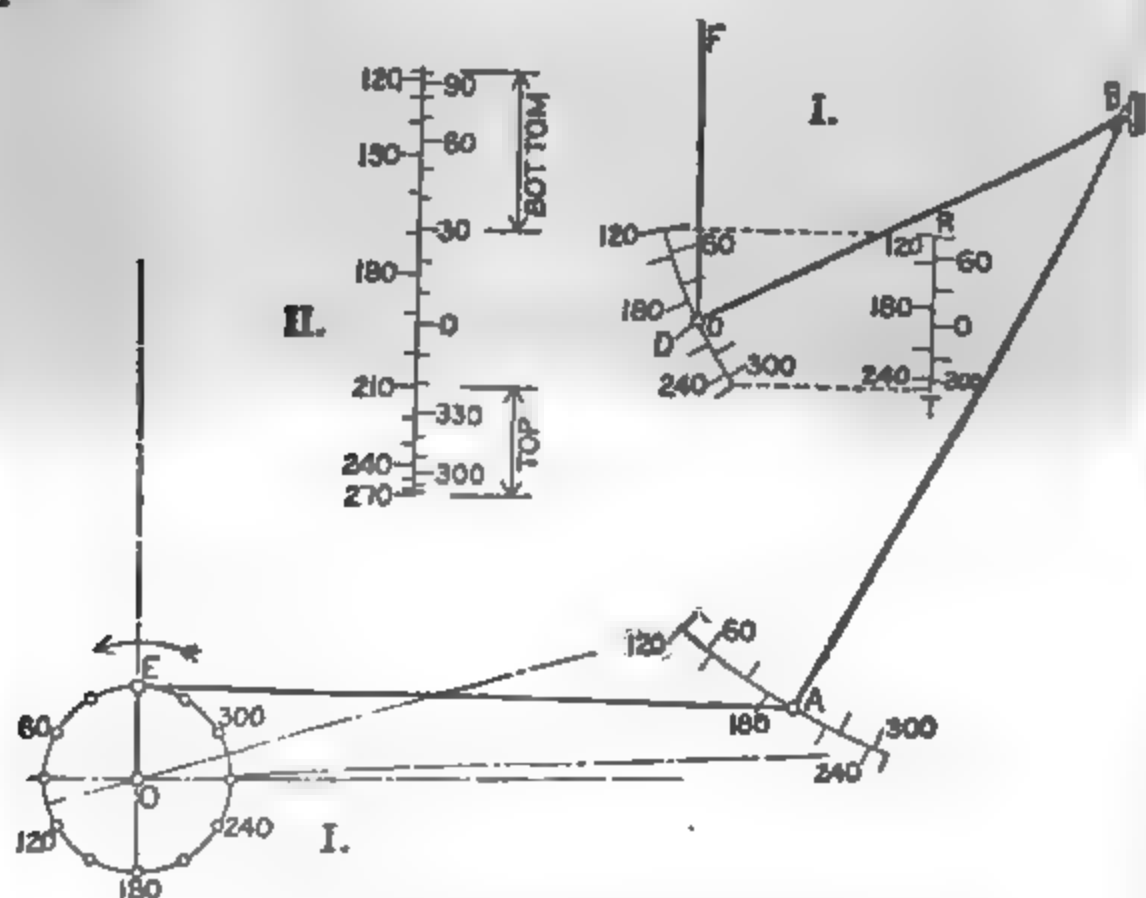


FIG. 497.—Movement of the Main Rocker-arm.

*Distortion of Eccentric-movement to Equalize Cut-offs.*—At first we shall proceed as if we did not know the proper setting of the eccentrics relative to the crank, plotting in Fig. 497 the movement of the main rocker ABD in terms of the eccentric-angle. This is a marked case of the condition in Fig. 363, and by pro-

ing the vertical displacements of the point D upon the line parallel to DF, and then enlarging this travel-scale at II., and that periods of opening having a considerable difference angular length can be made to begin at diametrically opposite positions of the eccentric. Thus we might use  $20^\circ$  to  $177^\circ$ ,  $200^\circ$  to  $51^\circ$ , with respective lengths of  $157^\circ$  and  $151^\circ$ : but these periods are longer than is needful for the main valve, and it seems better to use  $30^\circ$  to  $169^\circ$  and  $210^\circ$  to  $339^\circ$ , or  $139^\circ$  and  $129^\circ$  respectively for the bottom and top—remembering that we must have the greater opening in the bottom or crank end to equalize cut-offs against the effect of the connecting-rod.

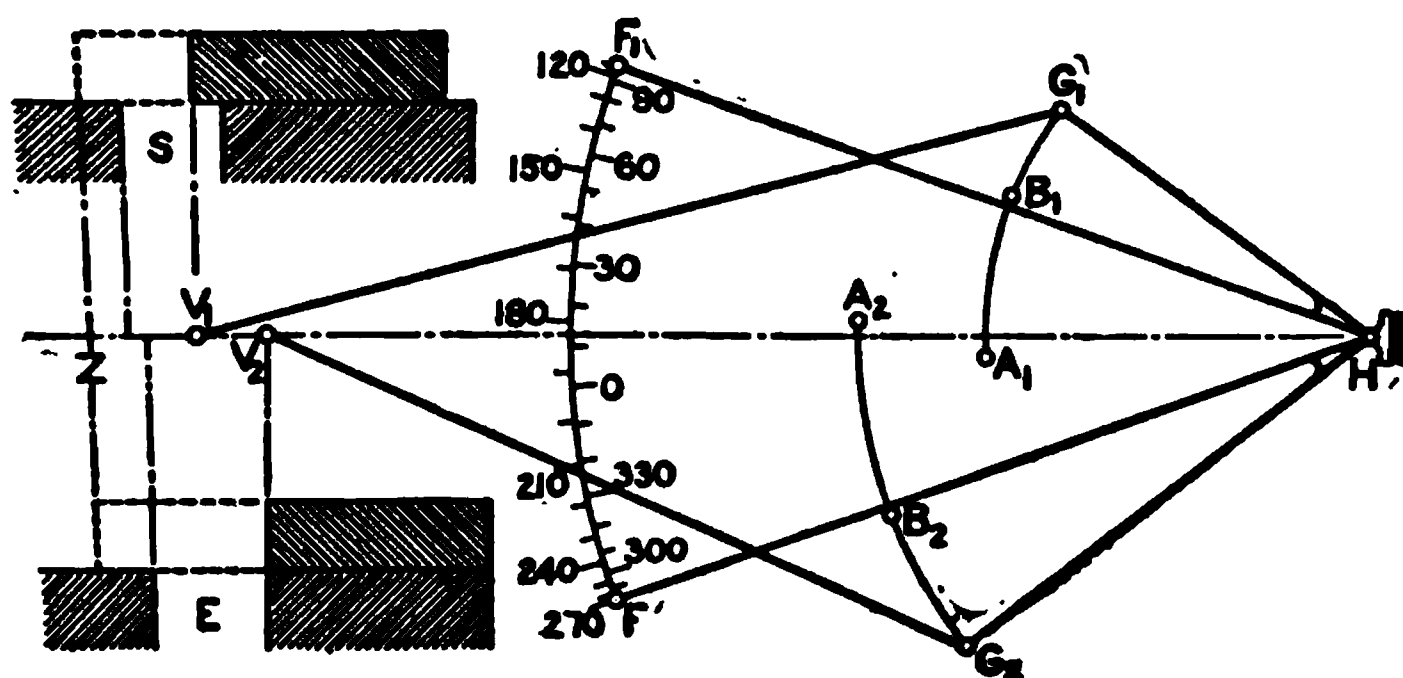


FIG. 498.—Bottom Valve-drives.

*The Bottom Valve-drives.*—The next step is to proportion the valve-drives for the bottom end of the cylinder, as in Fig. 498, having due regard to maximum lap of valve when closed, time of opening (marked B on path of G), and greatest width of opening.\* With the length of HG and GV fixed by approximate trials, we should finally determine the angle between HG and GF which will give HG movement over the proper range—everything being done, primarily, very much by the cut-and-try method. Note that HF is given the same swing up and down from the horizontal,  $F_1HG_1$  and  $F_2HG_2$ , being the extreme positions of this four-point rocker. At IV. in Fig. 496 are seen simultaneous positions of

\* It will be necessary to disregard subscripts intelligently in reading this discussion.

$G_1$  and  $G_2$ . The results got in Fig. 498 are, that the steam-valve is open from  $30^\circ$  to  $169^\circ$ , the exhaust-valve from  $190^\circ$  to  $330^\circ$  of the eccentric-travel as scaled on Fig. 497 L. Here, as in the next two figures also, the main arm HF is drawn half-size compared with the rest of the diagram.

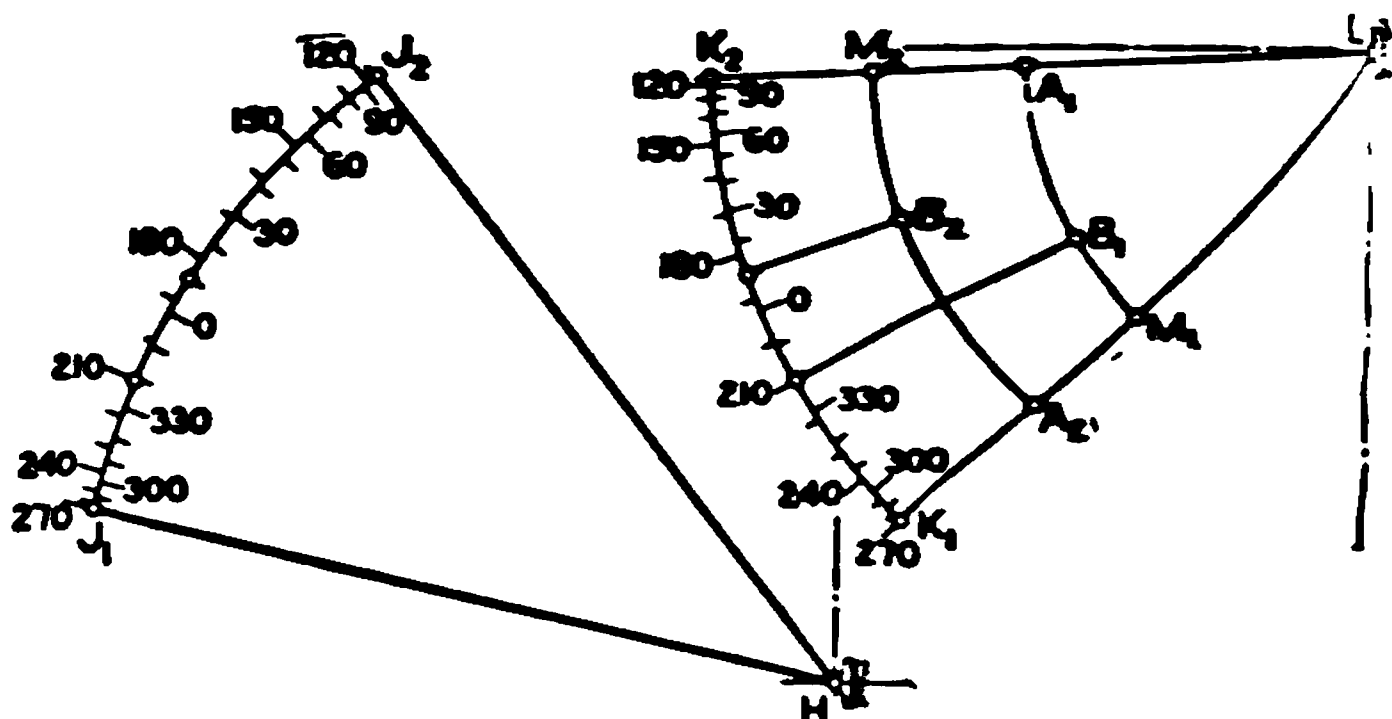


FIG. 499.—Adjusting the Movement of the Top Valves.

*The Top Valve-drives.*—Now comes a more difficult proposition. In Fig. 497 II. it appears that the vertical travel of the rod DF is much shorter for the period of "top" opening than for the "bottom." This was all right for the bottom valves, but must now be reversed for those at the top, if the movements of the two sets are to be alike. Fig. 499 shows how this result was obtained, after several trials, by swinging the movement-range of HJ up and that of LK down. The arcs  $M_1B_1A_1$ ,  $M_2B_2A_2$  are made just like  $G_1B_1A_1$ ,  $G_2B_2A_2$  on Fig. 498; and then the length of LK and the inclinations of the arcs  $J_1J_2$  and  $K_1K_2$  are juggled until a radius through  $B_1$  strikes at  $210^\circ$  on  $K_1K_2$ , and a radius through  $B_2$  at  $10^\circ$ —each diametrically opposite to the corresponding beginning of opening in the other end. For the preliminary trials of different proportions, points on  $J_1J_2$  can be projected horizontally to  $K_1K_2$ ; but after very nearly the right conditions are found, actual plotting must be done. The form of the top valve-drive is shown in Fig. 500, similar to Fig. 498, and giving almost exactly the same valve-action as to the three critical points.

(e) MOVEMENT OF THE CUT-OFF VALVES.—Inspection of Fig. 96 will show that the expansion valve can most simply perform its function if given a long “dwell” with its bars nearly over those of the valve-seat, and a quick movement to the right and back at the proper time. This action is secured by oscillating the driving-arms  $B'D'$ ,  $B'F'$  (on Fig. 496 I.) over an angle-range lying near the dead-point lines  $B'G'$ ,  $B'K'$ , each having, however, a much

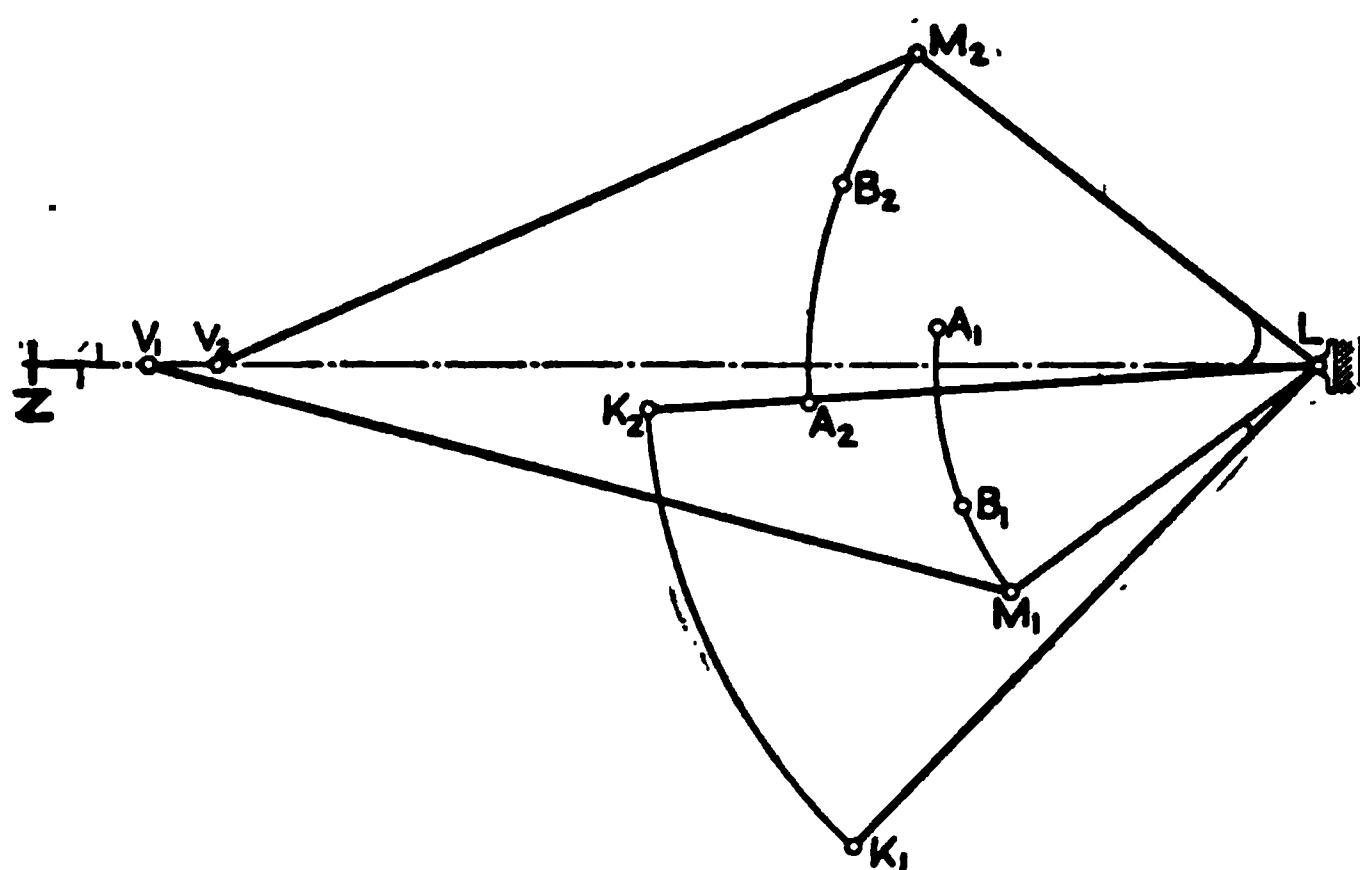


FIG. 500.—Top Valve-drives.

greater swing toward the side for opening. In Fig. 501 we develop a method for plotting accurately and at full size the movement of these valves, with a minimum of work and in a small space.

First of all, a small-scale drawing of the mechanism is made, partly shown by  $BdgH$  at I., to give the directions of the rods  $D'G'$ ,  $F'K'$  (Fig. 496), as at  $d_1g$ . Then taking the driving-arm  $B'A'$ , we use its half-size length to strike from  $B$  the arc  $A_1A_2$ ; next draw the eccentric-circle to the same scale on  $OE$ , divide it as marked, project the divisions horizontally to  $A_1A_2$ , and the divisions of this arc radially to  $D'D'$ ,  $F'F'$ , which are struck with the full lengths of the arms  $BD$ ,  $BF$ . These movement-scales are now transferred to the actual paths  $D_1D_2$ ,  $F_1F_2$ , and we are ready to plot the movement of  $G$  and  $K$ , proceeding as follows:

Draw the rod-slant line  $DG'$  for mid-position of  $D$ ; through  $D'$  pass an arc  $DS$  like  $GG_1$ ; for any position as  $D_1$  draw the line  $D_1RS$ , intermediate in slant between perpendiculars to  $D_1D'$  and  $DG'$  respectively; then  $DS$  will be the displacement of  $G_1$  from  $G$ , at the end of the arm  $HG$ , which is the same as the movement of

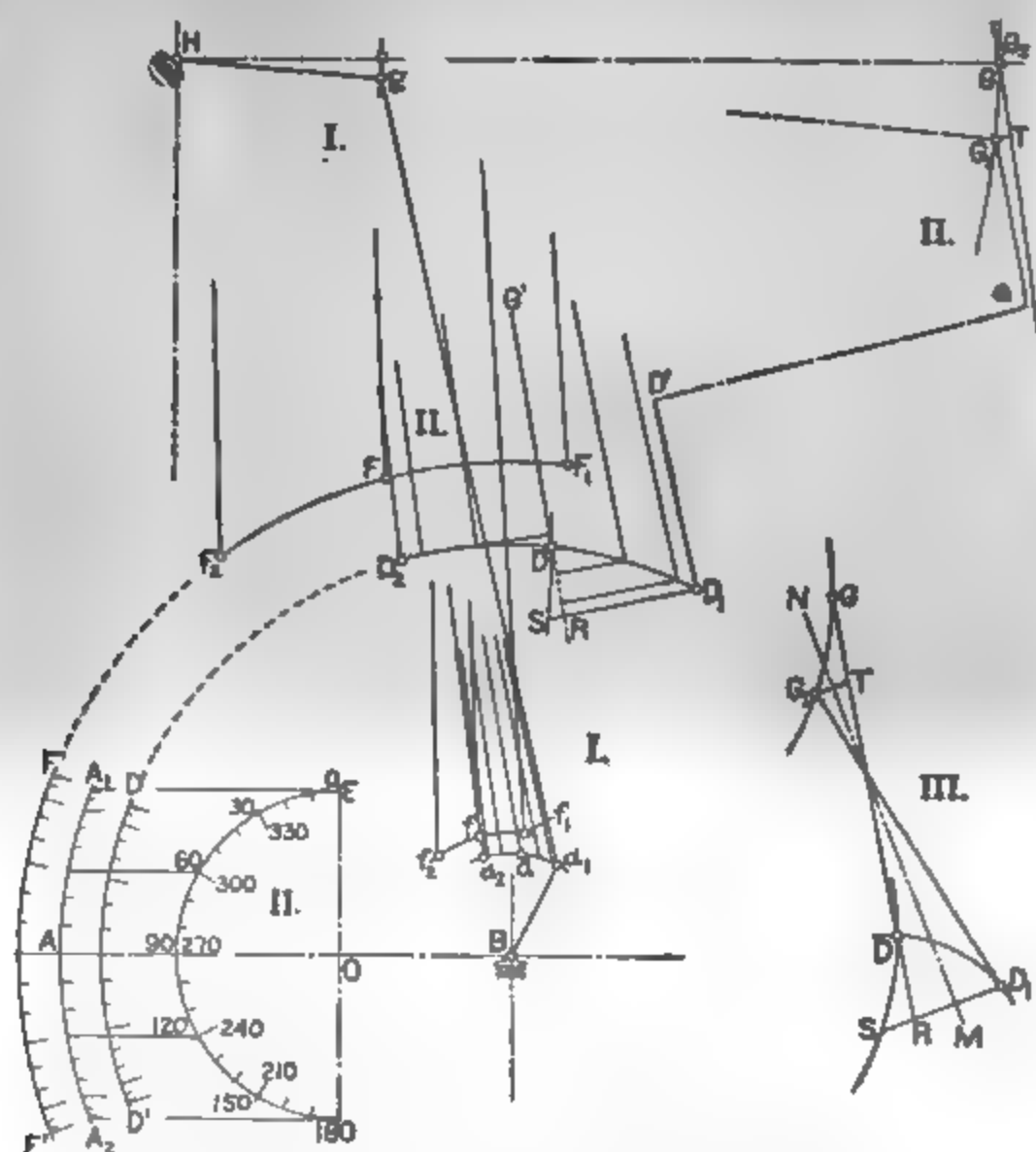


FIG. 501.—Action of the Cut-off Valve-gear.

the valve. To illustrate the equality, we draw  $GT$  parallel to  $DG'$ , transfer  $DR$  to  $GT$ , and draw  $TG_1$  parallel to  $RS$ . The principle is set forth at III. If  $MN$  is a line midway between the two rod-slants, and  $D_1R$  and  $G_1T$  are perpendicular to  $MN$ , we have  $TR$  equal to  $G_1D_1$ , which is the same as  $GD$ , wherefore  $DR$  and  $GT$  must be equal. On the drawing-board it is easy



give the  $D_1S$  lines the proper inclination, without a formal instruction such as will be involved in getting MN at III.

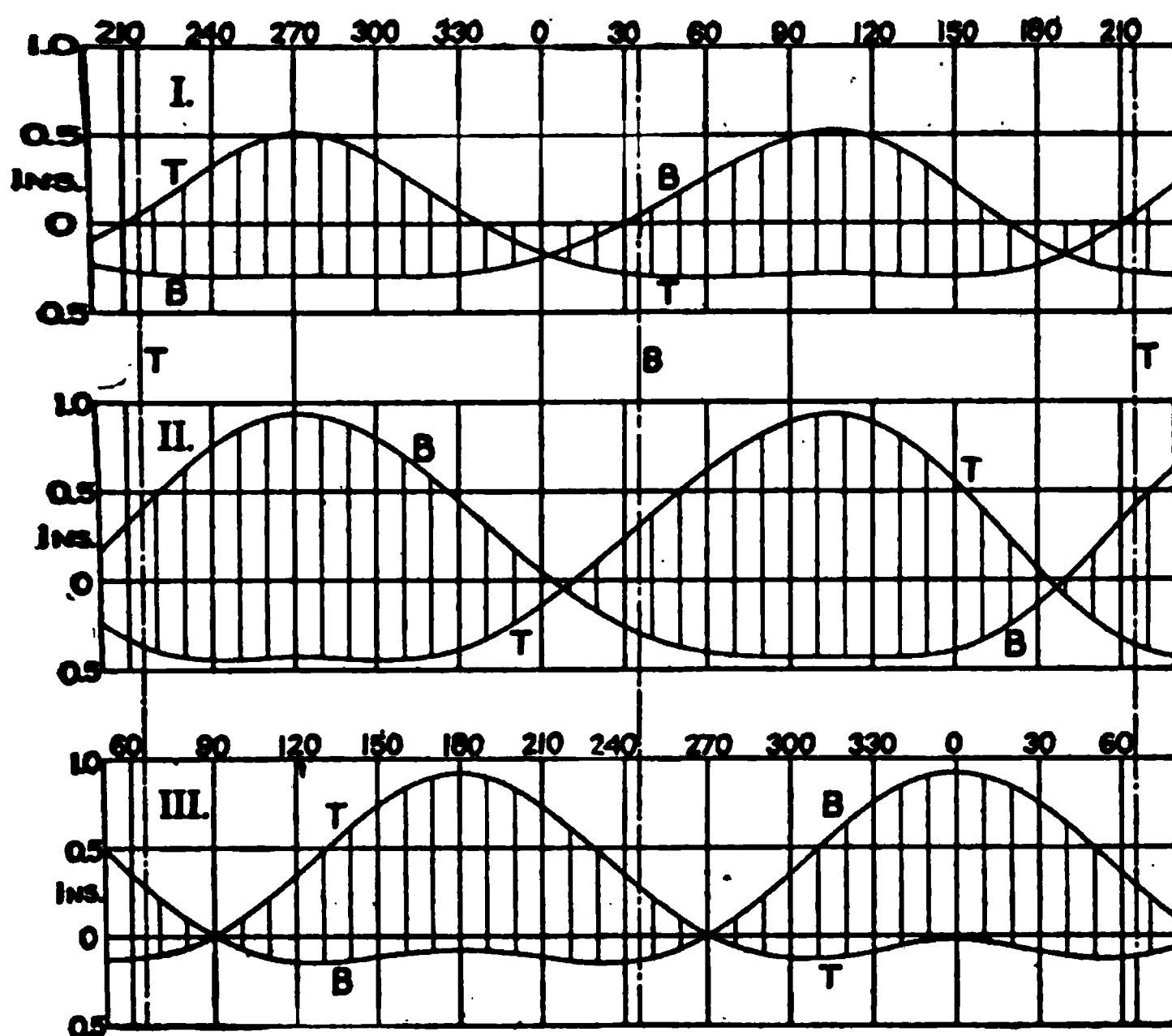


FIG. 502.—Primary Valve-movement Curves.

*Curves of Valve-movement* are plotted in Fig. 502, set I. for the main-steam, II. for the exhaust, and III. for the cut-off valves—all on the eccentric-angle base with zero at the top dead-center, and with the last set in about the proper relative position. On I. we note the difference of about ten degrees in the two main-valve cut-offs, the base-line representing the line-and-line or edge on edge position of the valve. Further, proper positions of the crank dead-centers are found, that for the top at  $215^\circ$ , that for the bottom at  $35^\circ$  of eccentric-travel. This gives the eccentric-setting in Fig. 496 VI.

The last step is to combine the two sets of steam-valve curves and see how well the variable cut-offs are equalized. The main

curves  $S_T$ ,  $S_B$ , and the piston curve  $P$ , are obvious. The letter  $J$  marks the apex of the top cut-off curve,  $H$  that of the bottom curve, at  $180^\circ$  and  $0^\circ$  of the expansion eccentric respectively. Where these curves cut the  $S$  curves, going up (valve moving toward right), cut-off takes place. The half-stroke cut-offs  $C_2$ ,  $C_3$  are exactly equal: at one-quarter (subscript 1) and at three-quarters (subscript 3) there are very slight irregularities. For the earliest cut-off (subscript 0), the actions are quite a little non-

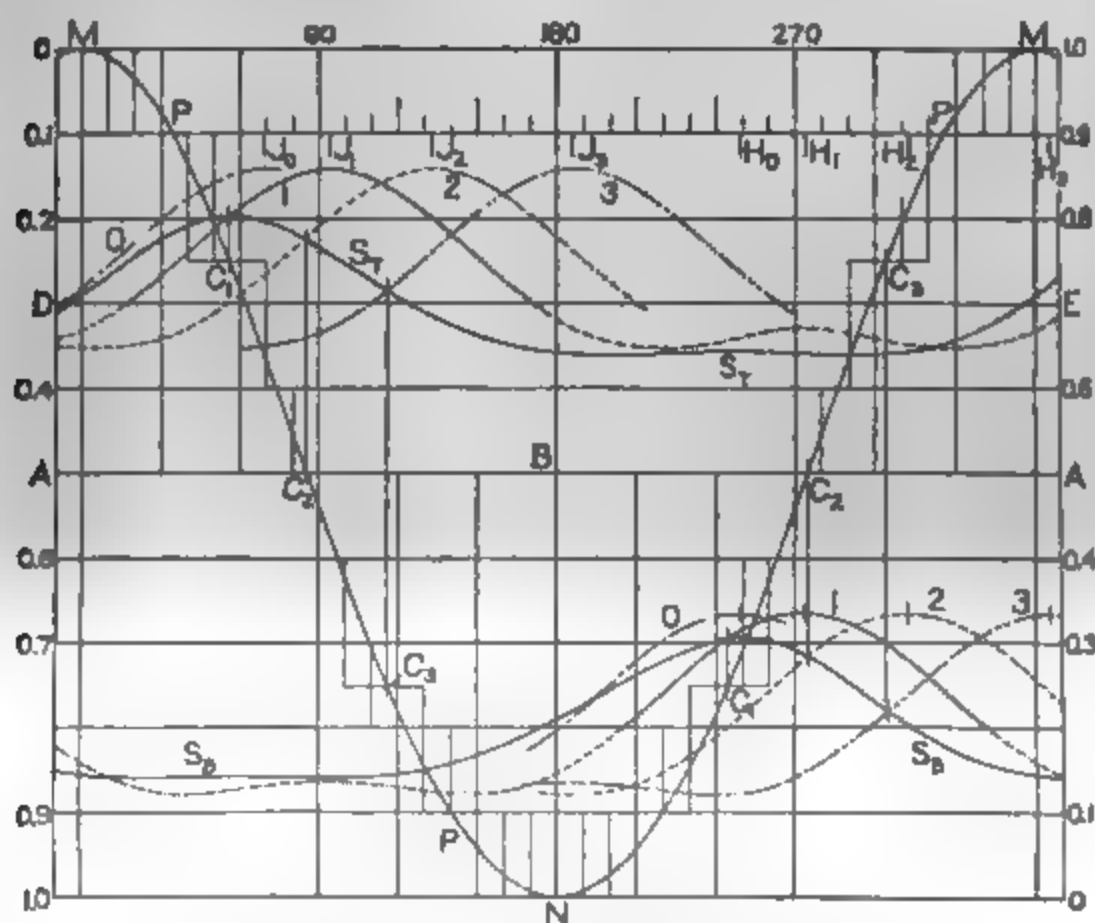


FIG. 503. Combined Curves of Valve-movement.

symmetrical. From the positions of the several  $J$ 's and  $H$ 's it is easy to trace out the eccentric-settings shown at Fig. 496 VL. Thus  $J_1$  shows that the crank will be at  $94^\circ$  when the eccentric is at  $180^\circ$ , or that the eccentric is  $86^\circ$  ahead of the crank: and the slant of the rod changes this to an actual angle of  $71^\circ$ .

These results do not represent quite the best that is possible with this gear. By carefully adjusting the several elements, even more symmetrical action can be got than is shown in Fig.

503. Enough has been given, however, to illustrate the possibilities of this type of mechanism and to set forth pretty clearly the methods to be followed in designing and discussing it. This appears to be about the ultimate case under the general idea of distorting from symmetry the simple harmonic motion given by the eccentric.

### § 59. Lift-valve Gears.

(a) DIFFERENT FORMS OF THE LIFT-VALVE.—The three typical forms of the valve which opens by lifting from the seat are shown in Fig. 504; all are double-seated, and are so arranged as to be

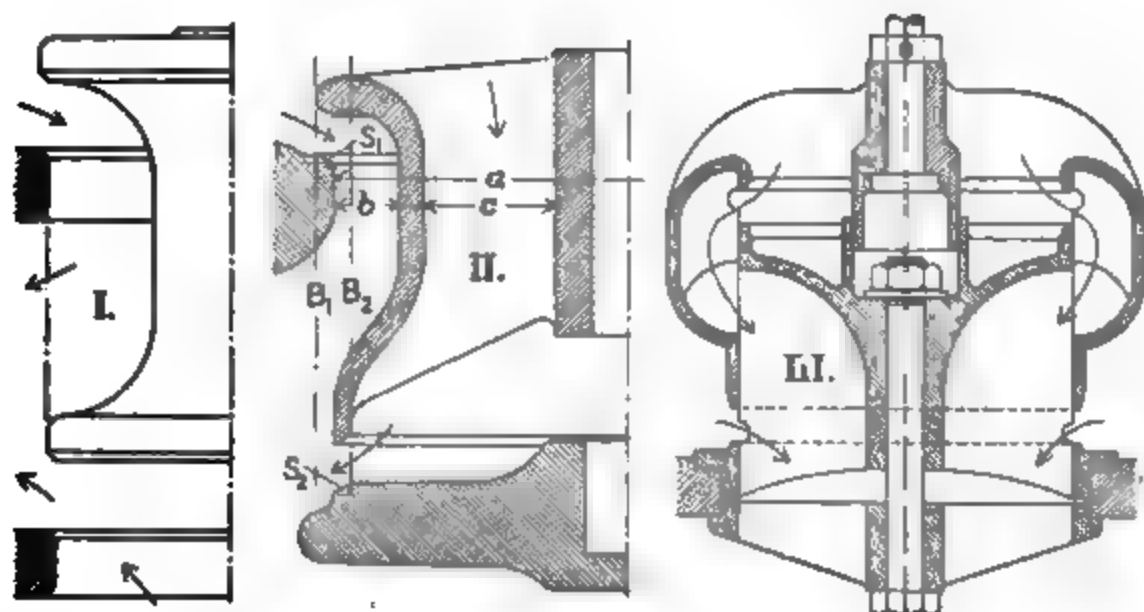


FIG. 504.—Different Types of the Two-seated Valve.—I. Poppet Valve, from Figs. 505 IV., 509; II. Hollow valve, from Fig. 267; III. Bell valve, as in I. and III., Fig. 505.

nearly balanced, only an annulus equal to or slightly exceeding the combined width of the two contact-surfaces being subjected to downward pressure when the valve is closed. The solid double-disk or poppet valve at I. is simplest in form, but requires more complication in the valve-chamber, because the steam coming to it must be given access to both top and bottom by means of passages formed in the casting: for a given diameter, however, it has a larger capacity than the valve which receives steam from the top only. Type number II. is the most usual valve in German

stationary practice, where the lift-valve so largely prevails. The dimensions marked on the figure suggest that the effective opening is determined by the diameter  $a$ , being the area of this circle less the cross-section of the valve-casting; and that the annular passages of widths  $b$  and  $c$  should be given equal areas, so as to permit equal flow from or to the two openings. Type III. is derived from II. by a kinematic inversion, the valve in one case corresponding in essential form with the seat in the other: it has the advantage that fitting the valve-seat into the cylinder is a much simpler matter than with the double-contact arrangement in Figs. 267, 508, and 509; but it appears to require rather more room, so that the volume of the cylinder-clearance will be greater.

These valves as well as the inserted valve-seats are made of hard cast iron. The contact-strips or seats are narrow, ranging, with the size of the valve, from one-eighth to one-half inch in width; they vary from a 45-degree cone as in I. to a plane surface as in III. An example showing a different direction of slant at the lower seat is given in Fig. 504 II. Almost always the valves are arranged to lift in opening, although engines have been built in which some of the valves open downward and are held up by springs.

Sometimes single-disk poppet valves are used in the low-pressure cylinders of pumping engines. Thus the triple engine whose diagram is given in Fig. 606 I. has these valves for exhaust on the L.P. cylinder; and the quadruple of Fig. 607, with eight sets of valves in all, has single-disk valves for the last three.

For some large engines four-seated poppet valves have been used successfully.

(b) VARIOUS LOCATIONS AND ARRANGEMENTS OF THE VALVES.—Fig. 267 is typical of best and most recent practice in horizontal engines. Quite often, however, the valves are placed at the side of the cylinder, either one above the other in pairs as in Fig. 505 I., all four in a row on one side, or two on each side. On a vertical engine the upper valves are best placed in the cylinder-head, as in II., but those at the bottom are nearly always on the side of the cylinder: the arrangement in Fig. 505 III. is used at both ends of the cylinder to which it belongs. The last example in

Fig. 505 is from the head-end of the cylinder, but the arrangement at the other end is just the same, except for the small difference in detail at the center, where a passage must be formed

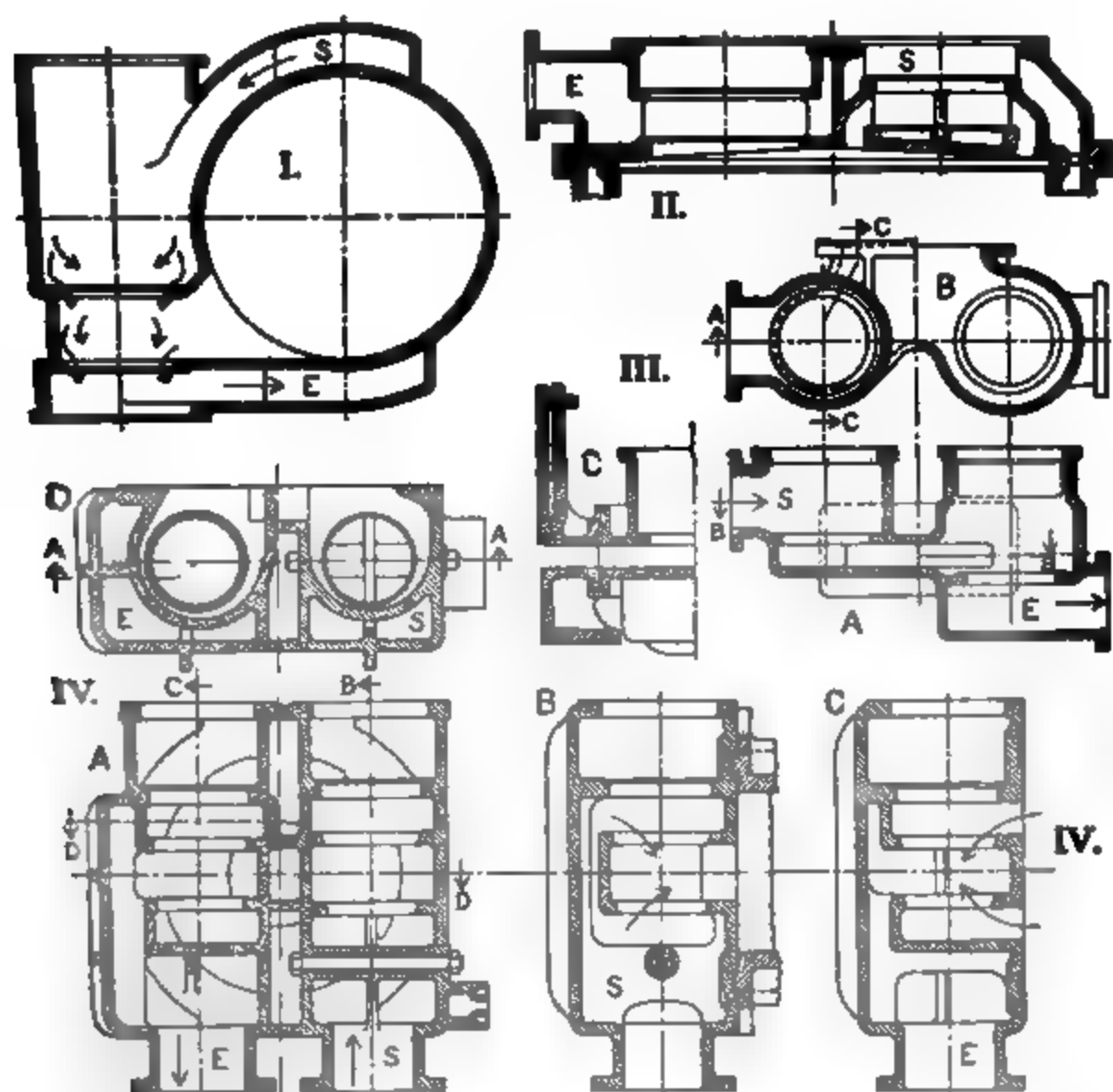


FIG. 505.—Different Arrangements of Lift-valves upon the Cylinder.—I. to III. German designs: I. Low-pressure cylinder with bell valves on same axis serving one port; II. Hollow valves in head of vertical high-pressure cylinder; III. Bell valves on side of vertical cylinder, placed side by side in a separate casting and serving one port; IV. Poppet valves in heads of high-pressure cylinders of large engines like Fig. 219, as built by the Allis-Chalmers Company for the New York Subway Power House.

for the piston-rod. These examples give a fair idea of the range of practice; for fuller information and detail the reader is referred to engineering periodicals or to Leist's *Steuerungen für Dampf-*

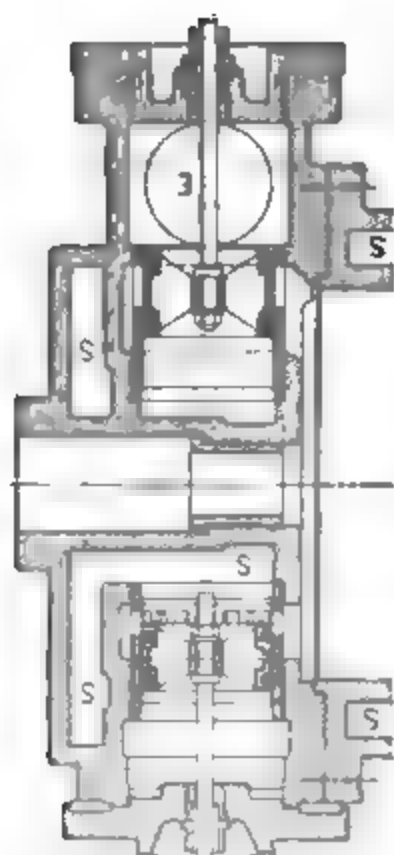


FIG. 506. Van den Kerchove Piston-lift Valves.

*Maschinen*—this remark applying also to the valve-gear discussion which follows.

A rather recent development along the line, shown as a novel design at the Paris Exposition of 1900, is the use of piston instead of seated valves in an arrangement otherwise like that which has long prevailed for lift-valves. From the point of view of one trained to the Corliss type for releasing gears, it is a decided advantage not to have to bring the valve to rest exactly at a certain level (against the seat) without shock. These piston-valves open at only one edge, but with the greater facility for absorbing their momentum quietly they can be lifted higher and dropped at greater velocity than the double-opening poppet valves. The external rigging is essentially the same

as for the latter type of valves.

(c) GENERAL ARRANGEMENT OF THE VALVE-GEAR.—With the valves located as in Fig. 267, the gear at the cylinder takes the form outlined in Fig. 507, the manner of driving the gear-shaft being indicated at II. Details of this particular gear are given in Figs. 508 and 510 III. As to the setting of the eccentrics, that for the steam valve must be only far enough ahead of the crank to take up the over-travel of the latch and give the desired lead when the crank is on dead-center. In Fig. 507 III. the crank is shown as if upon the valve-gear shaft and at dead-center, and  $\delta_1$  as marked measures

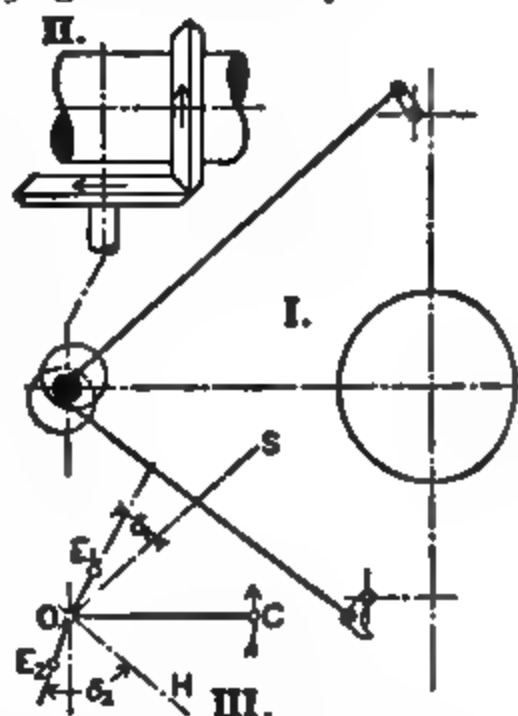


FIG. 507.—Outline of Gear at Cylinder; General Arrangement as in Fig. 267.

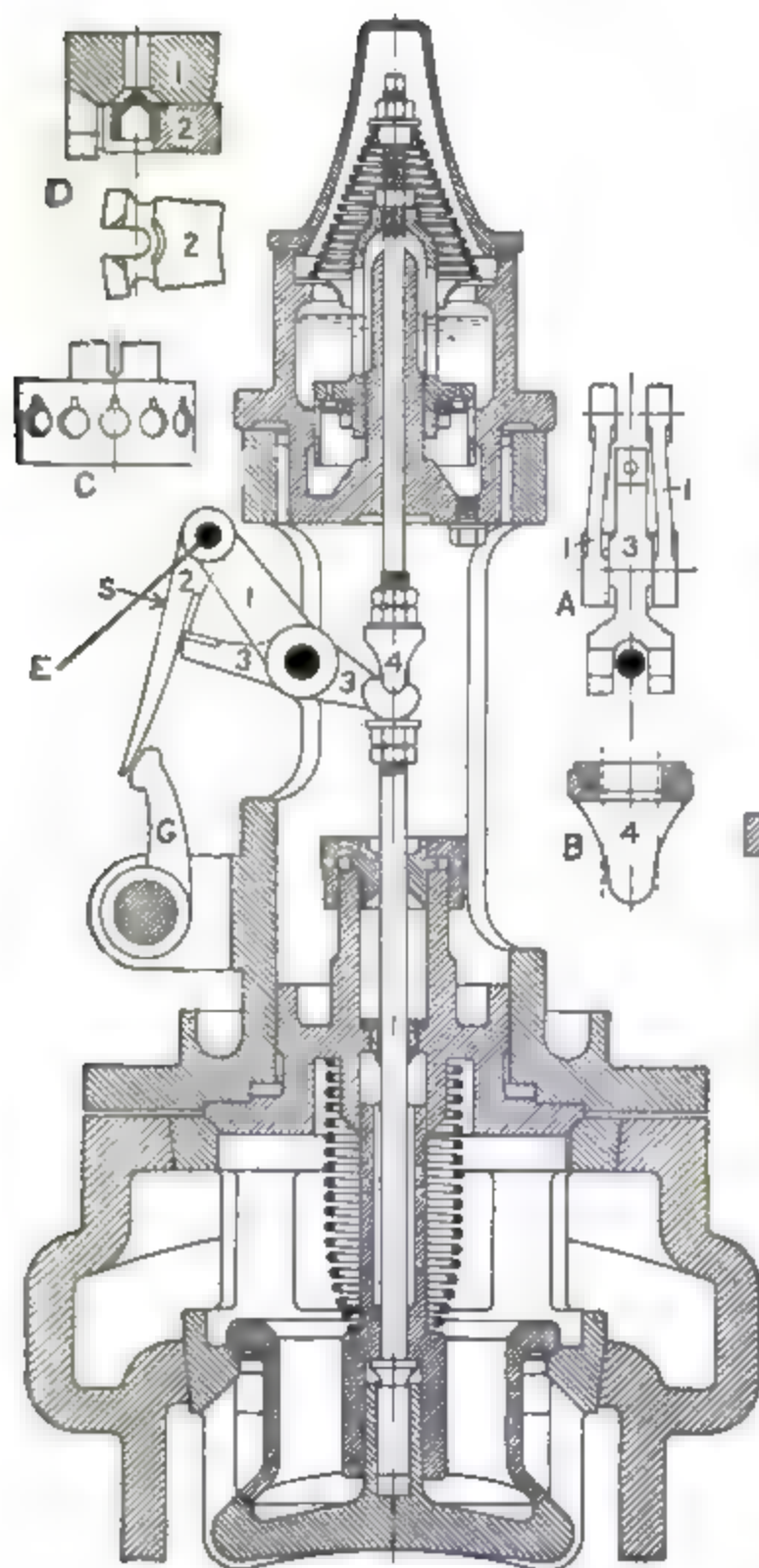


FIG. 508.—The Collman Releasing-gear, High-pressure Cylinder of Compound Engine. Scale 1 to 8.

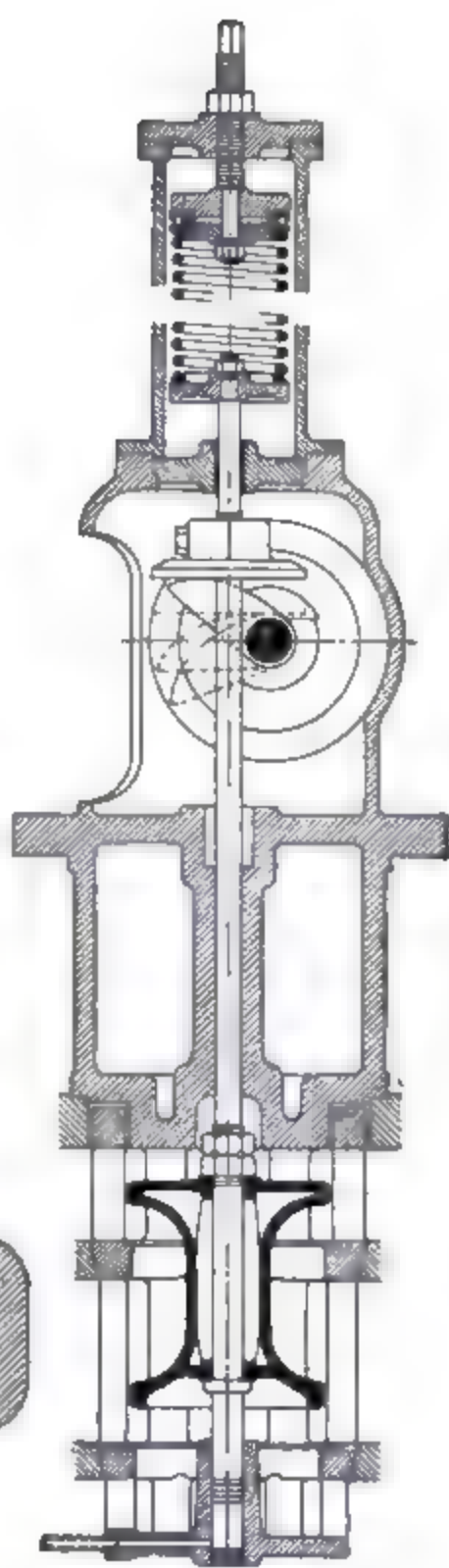


FIG. 509.—Detail of Steam-valve and Gear, 42'' x 60''. Cylinder shown in Fig. 506 IV. Scale 1 to 18.

the eccentric-angle, from the respective stroke-line OS. The exhaust-eccentric  $OE_2$  has the same angle of advance as for a Corliss engine—somewhere near 20 degrees.

(d) ADMISSION-VALVES WITH RELEASING-GEAR.—A first-rate German design is shown in Fig. 508. With the help of the detail at A, the arrangement of the driving-mechanism is clear; at B we see how piece 4 is made free to oscillate with the swing of 3. The forces which act to close the valve are the weight of all the parts attached to the valve-stem and the push of the lower spring, here placed in the steam-space. A rapid diminution of this force as the valve nears its seat is secured by the use of the lighter counter-spring at the top; and the movement is checked by an oil dash-pot, which has its plunger formed as better shown at C. The large holes let the oil pass freely while the valve is up, but only a small opening is effective when the valve is close to the seat. To avoid having the same oil-resistance at the beginning of the rise as at the end of the drop, a number of small check-valves are placed in a circle in the disk of the plunger, being made somewhat as suggested by the sketch at D. Air dash-pots are sometimes used, and it will be noted that the lower end of the valve-stem and guide on which the valve-hub slides will act as a small dash-pot with steam. As remarked under Fig. 267, to bring the drop-valve quietly and yet quickly to its seat is a more delicate task than the closing of the Corliss or other sliding cut-off valve.

In Fig. 509 the valve is raised by a rocking cam, carried on a spindle which is oscillated and released by the usual Corliss gear, with the dash-pot on the side of the cylinder. The cam is forged solid with the spindle, and is forked so as to bear under the collar on both sides of the valve-stem. Here the space under the lower end of the valve-stem is relieved to the atmosphere so that the steam may not get under it and tend to lift the valve.

(e) GEARS WITH ROCKING CAMS.—For the admission-valves of low-pressure cylinders (with constant, late cut-off) and for exhaust-valves in general, the desired movement can be very effectively obtained by means of some form of rocking cam or lever. Thus the exhaust-valves that go with Fig. 509 are lifted by a cam of the same shape; and the dotted outlines show how the body of



the cam swings down from the collar on the valve-stem after closure. This same idea, of letting the cams swing clear during valve-closure, is further illustrated in Fig. 510 I. and II. In lifting the valve receives a quick acceleration, then a steadier, rapid movement. Note the toggle-joint drive for the steam-valve in I. At III. we see a rotary oscillating cam, which has the same

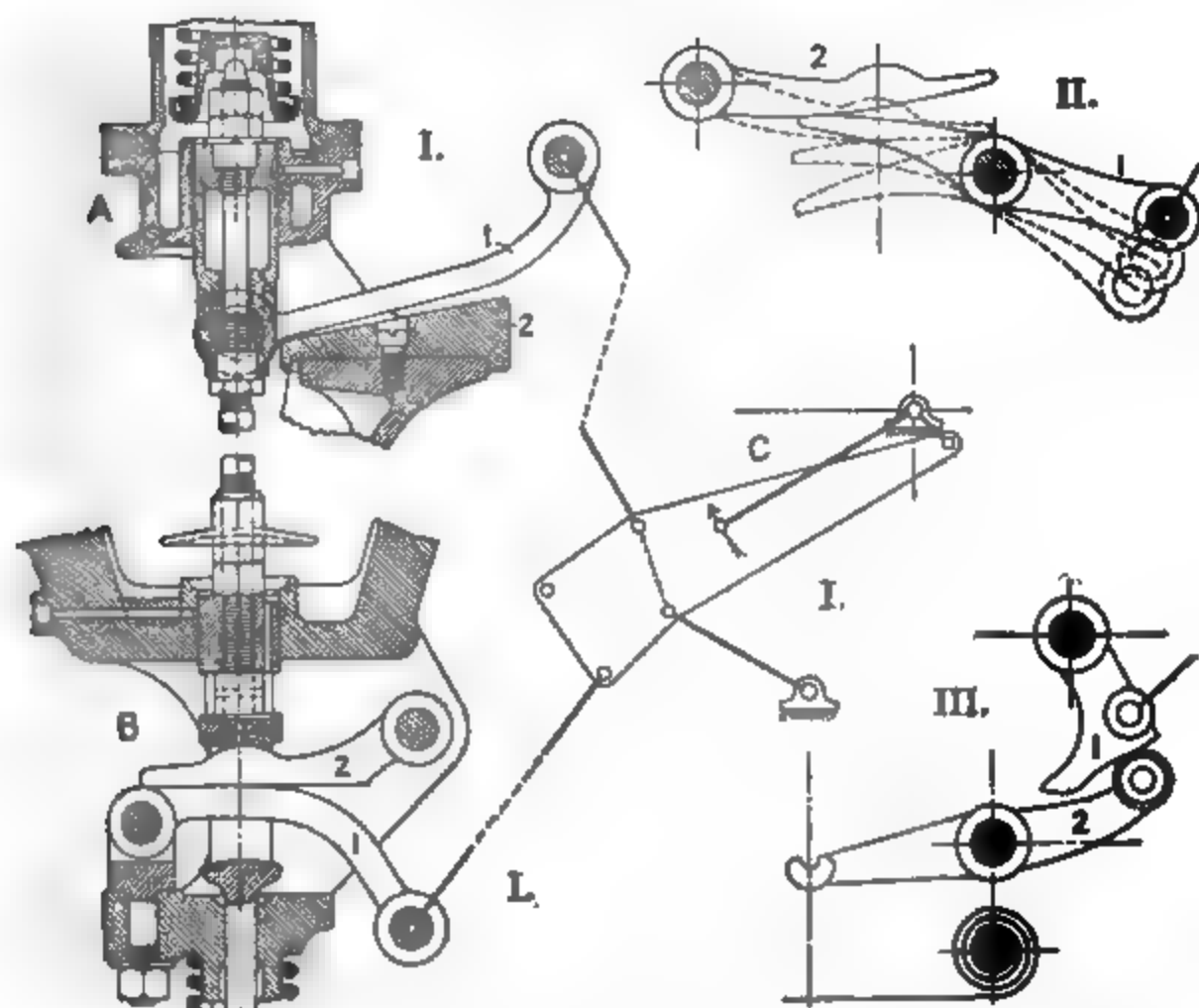


FIG. 510.—Rocking Levers and Cams.—I. Both valves of a low-pressure cylinder, with the eccentric drive; II. Another exhaust-valve gear; III. Oscillating cam and roller, with Figs. 507 and 508. Scale 1 to 12.

kind of movement as those which precede, but in the form and manner of action of its working-surface resembles the rotating cams in Fig. 511.

(f) **LINKAGE GEARS, WITH POSITIVE MOVEMENT.**—There are a number of designs, mostly German, of gears with the general arrangement in Fig. 507, but with a more or less complicated kinematic linkage between the eccentric and the valve, including

some scheme for varying the cut-off. These mechanisms show great ingenuity, the difficulty of the problem being much increased by the requirement that the valve must rest during closure, or that a movement-curve like that in Fig. 495 be secured. They are not, however, of sufficient importance to be given here the full illustration that would be required, and the reader is referred to Leist for fuller information.

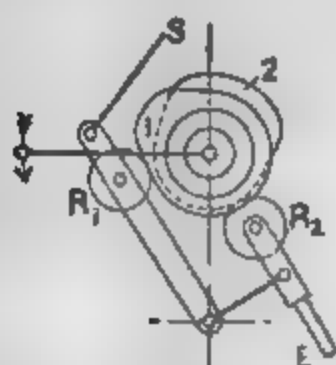


FIG. 511.—Cams on the Side-shaft.

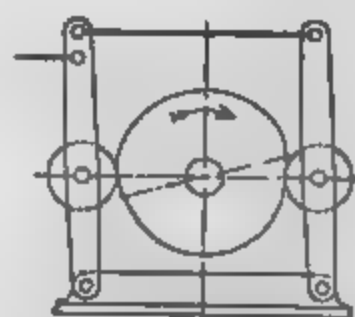


FIG. 512.—A Positive Cam-gear, Leavitt Pumping-engine.

(g) ROTATING CAMS.—Fig. 511 is typical of a large number of gears, mostly belonging to earlier practice and used at moderate speeds. With this device it is easy to secure quick movement in opening and closing, with periods of rest between the movements; but care must be taken not to form the profile so as to require too sharp accelerations where the movements begin and end. The valves are moved by single levers, such as would be formed in Fig. 508, for instance, by locking pieces 1 and 3 together and the springs, arranged as in Fig. 510 I., must be strong enough to carry the weight of all the rigging out to the rollers, besides furnishing the force needed to insure continual contact of the rollers on the cam.

A scheme for making the cam-gear positive, so as not to need weights or springs, is indicated in Fig. 512. This particular case, with two rollers bearing on one cam, can be used when the valve is to be open during half a revolution; for shorter periods, there must be two cams, side by side on the shaft, one designed so as to give the desired movement, the other then made to complement it and always fill the space between the two rollers. This gear

used, however, not with lift-valves, but with gridiron slides like those in Fig. 264.

To get variable cut-off with rotating cams, these are made in the form shown in Fig. 513, and best seen in the developed outline at B. In this case two cams are placed on the same sleeve, so as to reverse the engine; and levers 2 and 3, which work the valves at the two ends, are brought into the same plane, so as to be operated by the one set of cams. This gear, belonging to a hoisting engine, has the sleeve shifted endwise by hand; but in non-reversible engines the sleeve can be moved by a governor.

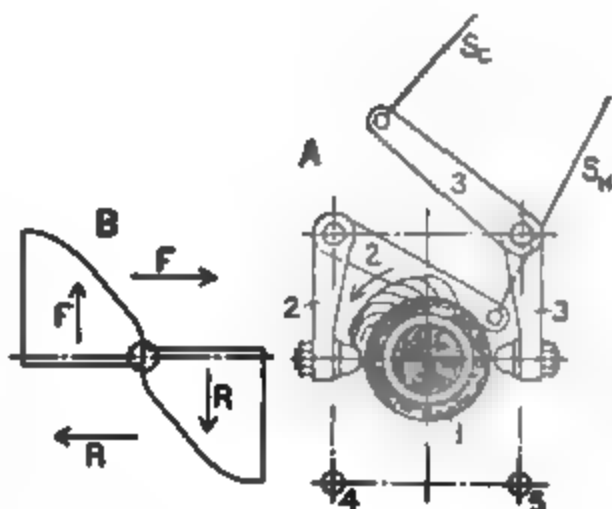
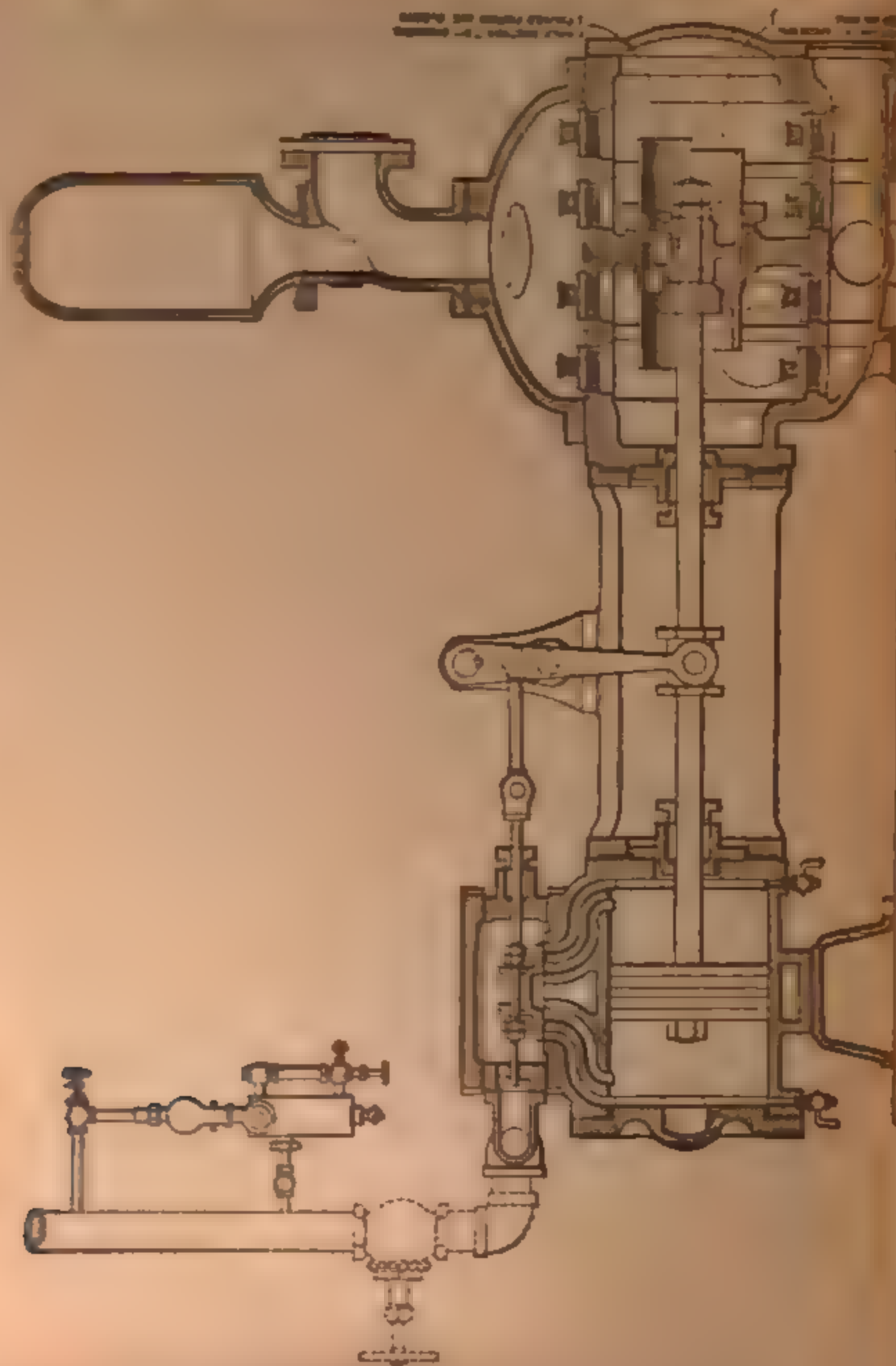


FIG. 513.—A Cam-gear which Varies the Cut-off and Reverses the Engine.

#### § 60. Valve-gears for Engines which have no Crank-shaft.

(a) THE DUPLEX STEAM-PUMP.—As to its valve-gear, this is the simplest engine that can be found under the above general title, because the valves are moved directly by the pistons, through the medium of plain oscillating levers which merely reduce the movement without changing its character: but to make this scheme work successfully, it is necessary to have each piston move the valve that controls the other cylinder. It will be noted from Fig. 514, and can be seen on Fig. 226, that one of the levers gives a direct movement to the valve, the other reverses the piston-motion—the two being proportioned so as to give the same ratio of reduction. That this is necessary can be most easily understood by a simple analysis of the movements.

In Fig. 515, A marks the right side, B the left side. For position 1., with piston A just reversing at left end of stroke, valve B (driven by arm A) is likewise at the left, port B is wide open at the right end, and piston B is moving from right to left; but in order that piston A shall presently move to the right, valve A



must move to the right ahead of it, wherefore valve A must have movement opposite to that of piston B. Following through the

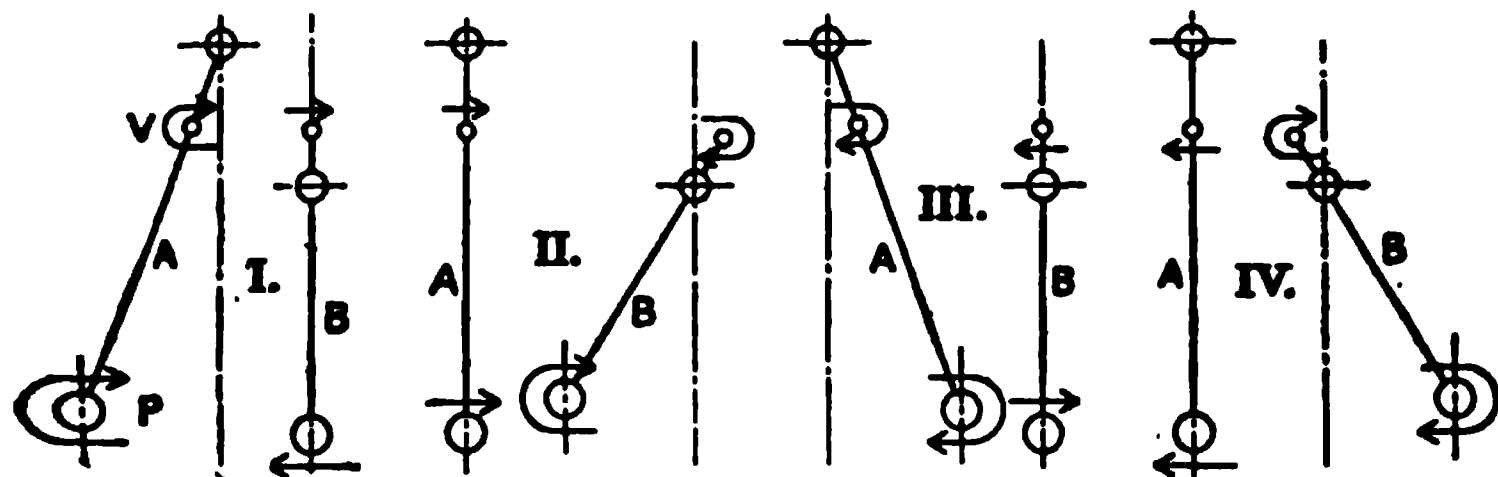


FIG. 515.—Valve-movement in the Duplex Pump.

Other three critical positions, and noting the directions of movement as indicated by arrows, we see that piston A leads, reaching any position half a stroke (one-quarter of the cycle) ahead of B.

Although moved by the pistons, the valves are not closely driven, but there is always some lost motion between the rock-levers and the valves. In Fig. 514 this clearance or back-lash is permitted at the nuts on the valve-rod, on each side of the valve; on larger pumps, as in Fig. 226, the same effect, with easier access, is provided for at the outer end of the rod. With positive connection, the valves would move too soon, so that the pistons could not make full-length strokes. Valve-setting in a pump like this consists in putting the pistons at mid-stroke and the valves in mid-position, and then making the rod-clearances the same on both sides; minor adjustments may be found necessary after the pump is started, the object being to secure equality in the two strokes.

Fig. 514—with confirmation from examples that follow—illustrates the fact that the steam-valves in engines of this class are made with zero lap. In this case we see also the use of two ports at each end, the outer one for live steam, the inner for exhaust. By thus having the exhaust-port open into the cylinder at some distance from the head, a cushioning effect is secured, independently of the

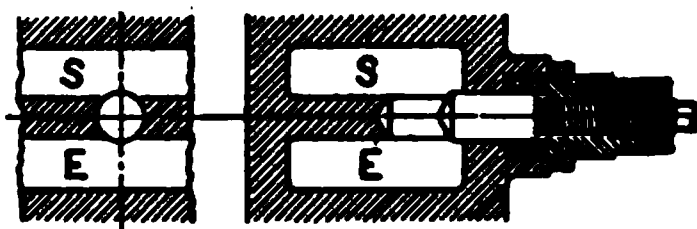


FIG. 516.—The "Dash-relief" Valve.

valve-action, which will prevent the piston from striking the cylinder-head. In order that this effect may be regulated at will (because less cushioning is needed at low than at high speeds), pumps of the larger sizes have stroke-regulating or cushion valves, arranged somewhat as shown by the sketch in Fig. 516. This valve varies the size of an opening through the wall between the two ports, permitting more or less exhaust through the outer part. Opening this valve lengthens the stroke, and *vice versa*.

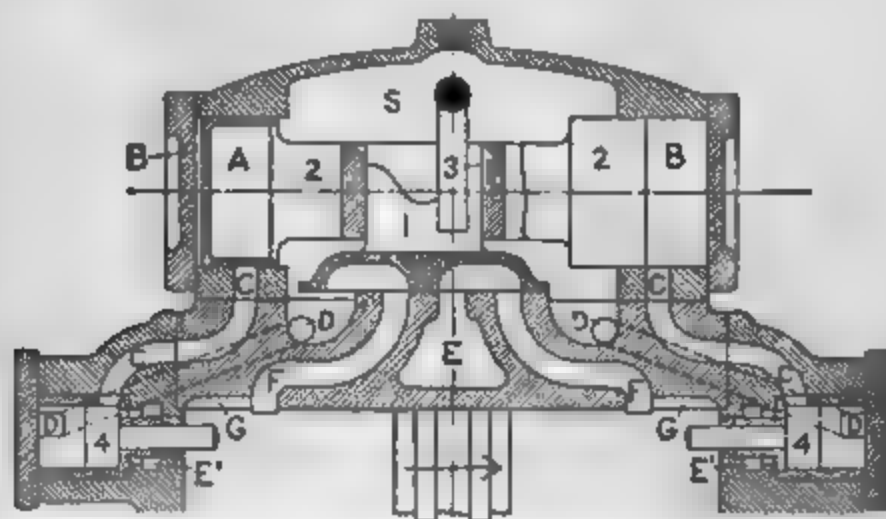


FIG. 517.—Valve-gear of the Cameron Simplex Pump.

(b) STEAM-ACTUATED VALVES — Duplex steam-pumps of all makes follow closely the original Worthington design; but single pumps show quite wide variations in their valve-arrangement. One typical example is given in Fig. 517, where the small primary controlling valves 4, 4, are moved directly by the piston as it comes to either end of its stroke. As in every case, the main valve 1 is actuated by a double plunger 2, which is controlled by variations of pressure in the end spaces B, B. Through the end of each hollow plunger is a small opening A, for the passage of full-pressure steam from the main steam-space S. At the inner side of each valve 4 is a passage E' which leads to the main exhaust-port E, and these valves are kept shut (pushed in, toward the cylinder) by the full steam-chest pressure, continually admitted through the ports D, D. Suppose now that the piston, coming to the right end of the stroke, pushes out valve 4; there is an imme-

the exhaust from the right end-space B, the steam escaping much more rapidly than it can get in through A; the result is that 2 is quickly pushed toward the right, but with a movement that is checked both by cushioning in front and by a drop of pressure in the left end-space B. Through the openings A, A, pressures are soon equalized, and the plunger is held fast by the friction of the valve until the piston gets to the other end of the cylinder. Note how far in from the cylinder-heads the openings of the main ports F, F, are placed, in order to insure a strong cushioning effect: the small grooves G, G, are not sufficient to mar the cushion, but make it certain that the piston shall not get stuck when just clear of the stem of valve 4. The starting-lever 3 is worked by an external handle, and can be used for moving the valve when starting the pump if it does not move off freely without help.

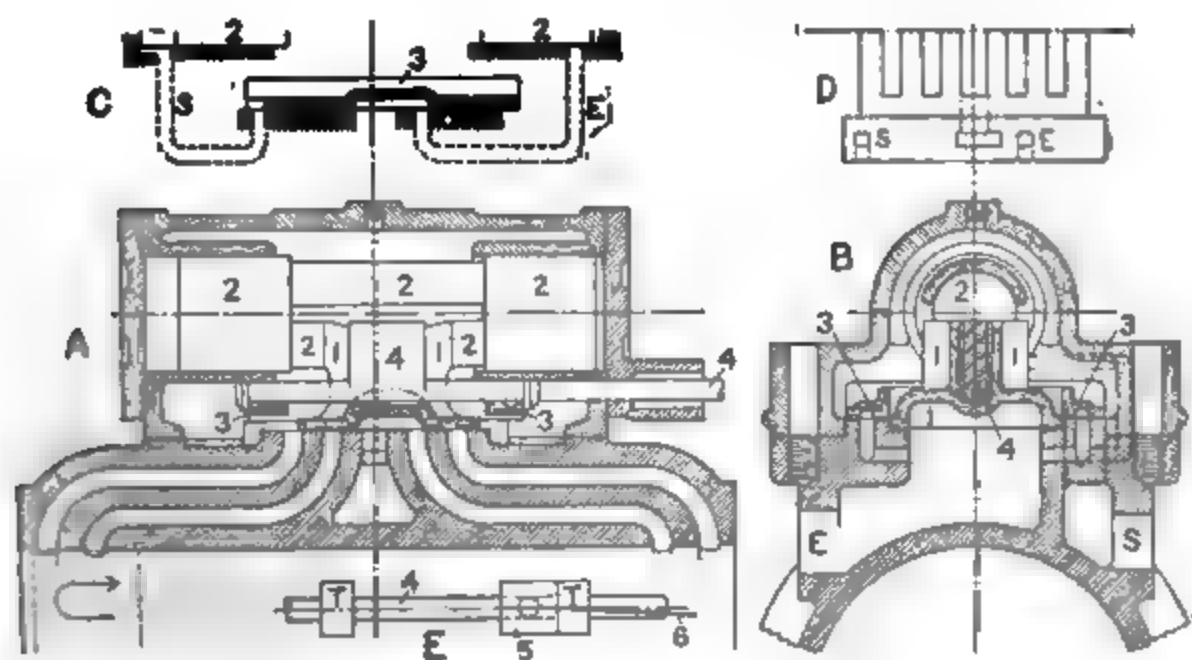


FIG. 518.—Valve-gear of the Deane Pump.

A majority of the valve-gears of this class is represented by Fig. 518, where there is a light auxiliary slide-valve 3—in shape a hollow rectangle surrounding the main valve—which controls ports leading to and from the spaces at the ends of 2. A plan of half the valve-seat is given at D, the other half being just the same, but with the small ports S and E reversed end-for-end. The arrangement of these ports is diagrammed at C, where the





hole valve-gear is carried upon the top cylinder-head, so that the controlling-valve 6 can be moved by the light rod 9 and is given a slight movement as the piston nears each end of the stroke. The several ports, T for the top end, L for the bottom end, and E for exhaust, are carried through the head-casting and down the side of the cylinder as necessary.

The lengthwise section, view A, shows the main valve 1 and the controlling slide, with two unequal pistons 2 and 3. By the action of the secondary valve 6, steam is alternately admitted to and exhausted from the space at the right of piston 3; the space at the left of 2 is always open to the exhaust, through the small passage K, shown in view D and partly indicated on C. The diameters are in such a ratio that the area of piston 2 is one-half that of 3, which gives the same driving-force in both directions.

The arrangement of the small ports from valve 6 can best be seen on the outside view of the main valve-bushing, at C: H is the steam-port, carried nearly to the end of the casing, and on the inside extended by a notch, as shown at D; G is the exhaust-port, stopped short so as to insure cushioning, after the usual manner; and F is a small passage leading to the main exhaust. Valve 6, rounded to fit bushing 7, is in effect a common slide-valve (not balanced), and is surrounded by steam of full pressure. Raising this valve (as in the drawing) admits steam back of piston 3, pushes valve 1 to the left, opens the main port T, and drives the engine-piston down. Lowering valve 6 shuts off steam from port H and connects G to F, whereupon the excess of internal pressure upon 3 moves the slide to the right, and prepares for the up-stroke of the main piston.

(d) INDICATOR DIAGRAMS FROM STEAM-PUMPS.—The duplex pump gives, when working properly, a steam-diagram that is almost a perfect rectangle, corresponding in general shape with the rectangular diagram that can be taken from the water end. The example at I. in Fig. 520 is from a compound pump, similar to Fig. 226. To find the driving-force, the effective pressures would have to be reduced and combined as in Fig. 144. Since the mass of the moving parts is insignificant, it is essential that the driving

steam-force shall always keep close to the resisting water-force, if the motion is to be smooth.

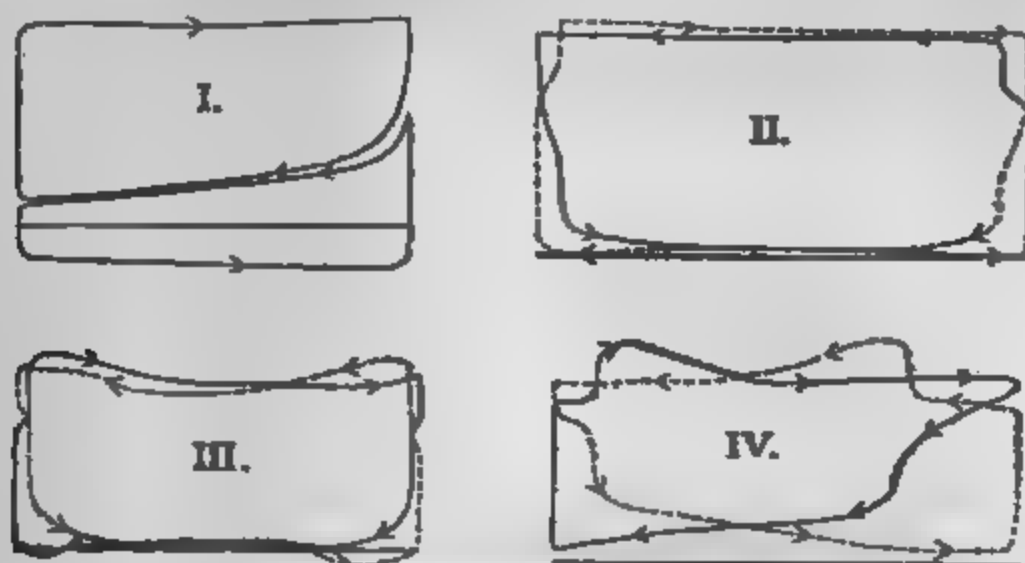


FIG. 520.—Indicator Diagrams from Steam-pumps.

The remaining diagrams in Fig. 520 are from single pumps; and in all of them is shown a peculiar action due to the inertia of a long column of water in the discharge-pipe. While the pump is pausing, at the end of the stroke, the water keeps on moving, and gets away from the water-plunger, at least to the extent of greatly decreasing the reaction upon this plunger; then when steam is admitted, the piston gives a little jump until checked by the water. As soon as it is thus checked, the entering steam has a chance to rise to full pressure and the escaping steam to fall to (or toward) exhaust pressure, and the real working stroke begins. This action is most marked in IV., where, at the right end, the forward pressure (dotted) is at first less than the back-pressure, making it look as if the plunger was, through a short distance, dragged or sucked after the water. Examination of any of these diagrams shows that after the full driving-force is established, it remains nearly constant throughout the stroke. The peculiar exhaust-lines in III. are due to the fact that the exhaust-pipe of this mine-pump discharges into the water of the "sump" or suction-pit.

With these water-velocity effects minimized, the single pump tends to give a simple rectangular steam-diagram. It must be

understood that the air-chamber on the discharge-pipe has a large share in the hydraulic force-action above described.

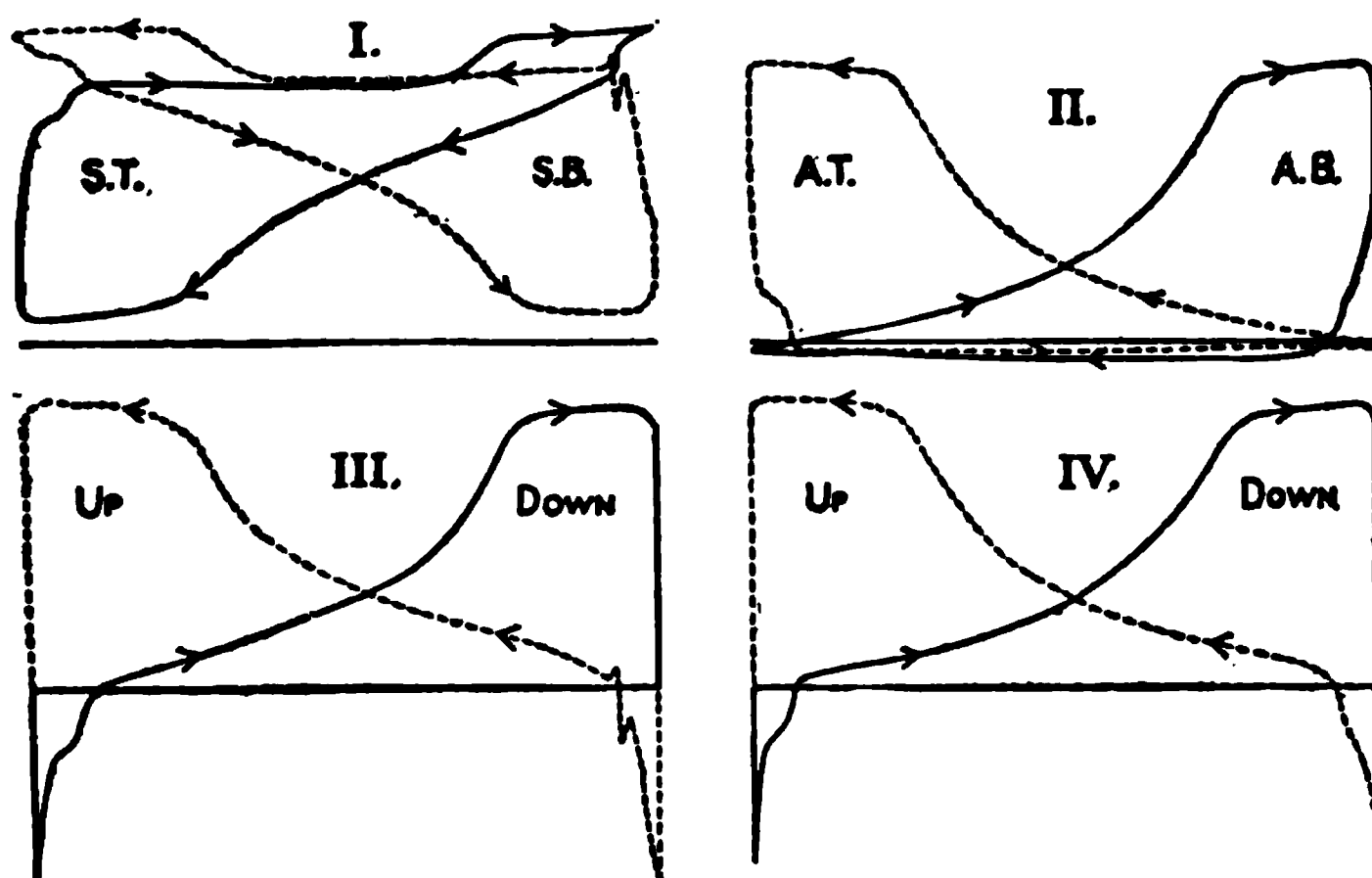


FIG. 521.—Pressure Diagrams from the Air-brake Pump.

When the resistance is not constant, but increases along the stroke, the steam-diagram takes the peculiar form shown in Fig. 521, where I. and II. are corresponding diagrams from steam-end and air-end, brought to the same scale of pressures. Considering simultaneous lines of forward-pressure and back-pressure in I., we see that a small driving-force at first is secured chiefly by throttling the exhaust; instead of taking place while the piston is at or near the end of its stroke, the release (with drop in pressure) is distributed along a good part of the return-stroke. As the air-resistance increases, the piston, which started off rapidly, slows down; and during the expulsion of the compressed air the two steam-pressures are nearly constant at their maximum and minimum values. Diagrams of effective driving-force and effective resistance are plotted at III. and IV., from I. and II. respectively, to show how very closely the two force-actions agree, there being between them only the small differences needed to accelerate the pistons.

This action is characteristic also of the low-pressure compressors

used as "air-pumps" on condensers. An essential point in the proportioning of the valve-gear is the use of very small ports. In Fig. 519, for instance, with a piston  $9\frac{1}{2}$ " in diameter, the port openings through the valve-bushing are only  $\frac{3}{8}$ " by  $1\frac{1}{2}$ "; or the ratio of port-area to piston-area is about 1 to 130, as against perhaps 1 to 10 in an ordinary engine. It need hardly be remarked that such conditions of working as are shown in Fig. 521 do not tend toward economy of steam.

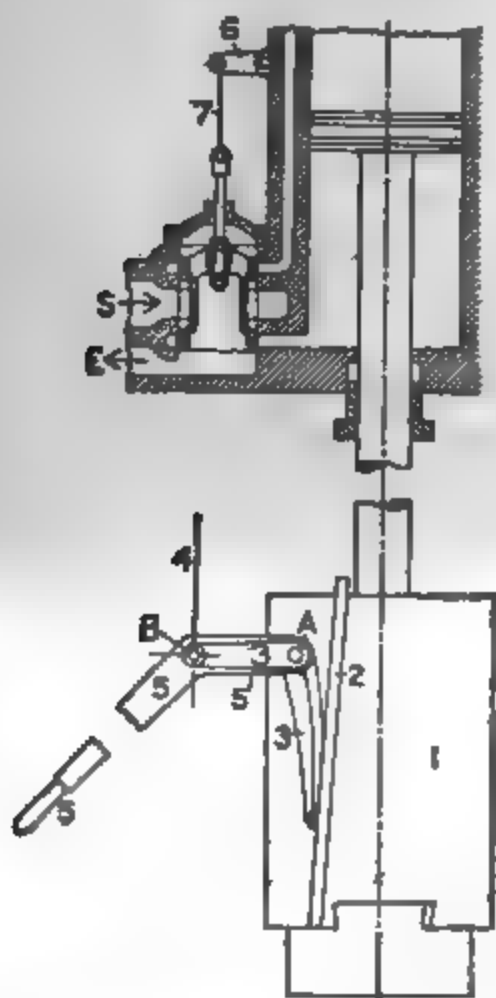


FIG. 522.—Valve-gear of Bement and Miles Steam-hammer.

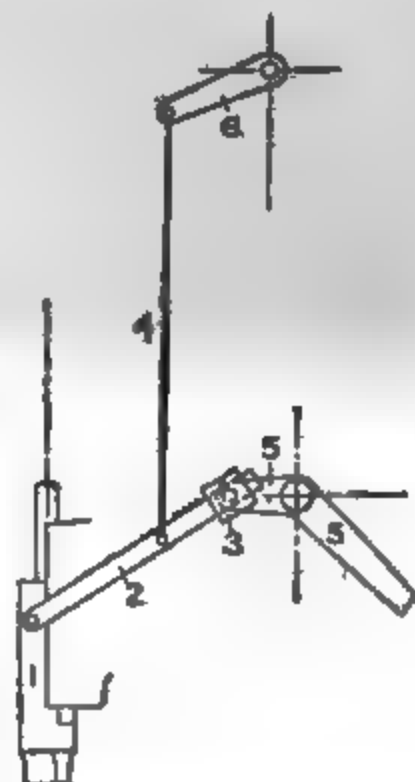


FIG. 523.—Steam-hammer Valve-gear with Oscillating Lever.

(c) VALVE-GEARS FOR STEAM-HAMMERS. — In this type of machine, with a very heavy reciprocating mass, it is entirely possible to drive the valve from the main piston: but with the automatic movement is combined an element of hand-control, which is in more or less continual use during the operation of the

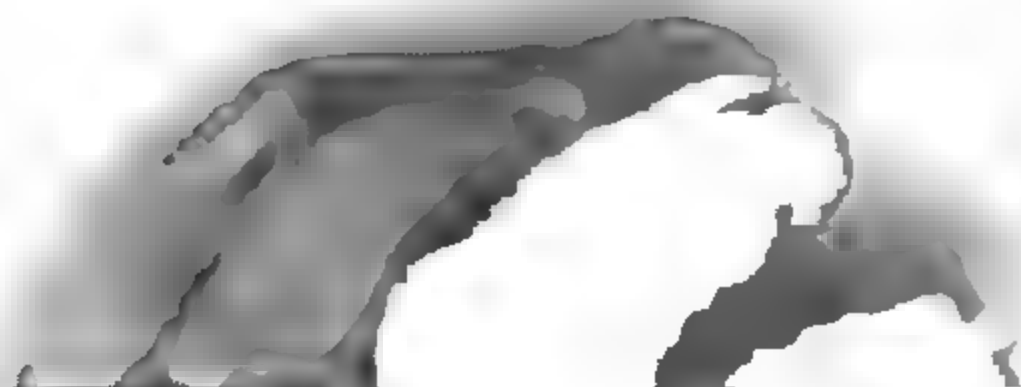
hammer. In Fig. 522, for instance, the hammer-head 1 carries an inclined slide-bar 2, which acts upon the cam arm of the bell-crank rocker 3, so as to oscillate this rocker as the hammer moves up and down. Through the rod 4 and rocker 6 (rod 4 takes hold of an arm on 6) the valve is given a considerable movement, but not enough for full-stroke working of the hammer. To get a piston-movement of variable length and intensity, hand-control through the lever 5 is added: this lever is pivoted on the frame at B, and carries at A the pivot of rocker 3. Simply to change the fixed position of 5 has the effect of changing the length of the valve-rod, raising or lowering the mid-position of the valve. Moving 5 with each blow supplements the automatic component of the valve-movement. It can easily be seen that raising the handle will lift the hammer and vice versa. In the figure, everything is drawn in mid-position. The valve must have some lap, not be made like the pump-valves just discussed; but this figure is only a sketch, intended to show form rather than proportions.

It is obvious that a machine of this class will be subjected to severe shocks, so that as little as possible of the valve-gear should be fastened to the hammer. Some designs, however, have an oscillating lever jointed at one end to the hammer-head, as in Fig. 523. In this case the valve, actuated by arm 6, is a D valve in effect but oscillates on a curved seat. Both the hammers here shown are double-acting, receiving steam on the top of the piston to increase the force of the blow: but many large machines use the steam for lifting only. Always, the operator has one hand on the main-valve lever, the other on the throttle-valve, so as to regulate the supply of steam as well as the working of the engine-valve.

(f) THE SELF-CENTERING VALVE-GEAR.—In certain regulating or controlling devices, such as reversing-gears for large engines and the like, it is desired that the piston shall follow a primary controlling lever, coming to rest at a point in its stroke similar to the position of this lever on its range of movement. Several devices of this sort are illustrated and described in § 65. Briefly stated, the principle of these gears is as follows:

The slide-valve is normally at mid-position, the piston being

at 1000. To move the piston, the valve is drawn to one side by moving the hand-lever, and as the piston advances it operates a secondary mechanism which brings the valve back to its position. The further the hand-lever is displaced, the further must the piston move in order to center the valve and again bring back to 1000.



## CHAPTER X.

### GOVERNORS OR REGULATORS.

#### § 61. Types of Rotary Governors.

(a) GENERAL CLASSIFICATION. — The two common types of speed-regulators have already been described as to general form and operation, the fly-ball governor in § 56, the shaft-governor in § 2 and 48. The essential difference is, that in the first type the governor-mechanism has its working movement in a plane that passes through the axis of rotation and turns about that axis; while in the second type the mechanism works in the plane of rotation, this plane being perpendicular to the axis of the shaft. When the fly-ball governor is operated wholly by centrifugal force, because the inertia-forces due to a change in the speed of rotation, being perpendicular to the plane of the mechanism, can have no component to influence the movement of the latter; but in the shaft governor this kind of inertia may play an important part. Differences in the way that the governor takes hold of and controls the valve-gear are of less importance than might at first sight appear.

(b) FUNCTIONS OF THE GOVERNOR.—These can be most clearly set forth by stating that in the study of the action of the governor two main questions are to be considered. The first is the question of "regulation", or of the manner in which the speed in steady running varies with the load; the second is concerned with "adjustment", or with the behavior of the governor while in the act of accommodating the engine to a change of load: involved in both, and by no means of subsidiary importance, is the question whether the governor will hold steadily the position corresponding

to a constant load, without yielding unduly to the action of secondary disturbing forces. By "close regulation" is meant that the whole range of load is covered with only a small change in speed—this change being normally a decrease as the load increases. An ideal governor, while steady under constant load, would respond at once to any change in the main controlling forces, following the load to the new position of equilibrium and stopping there, without superfluous movement on its own account. It is desirable that the actual governor shall regulate as closely as is consistent with steadiness, and shall adjust quickly and positively; but generalizations as to realized or possible performance will be deferred until we have made a pretty close analysis of several typical governors.

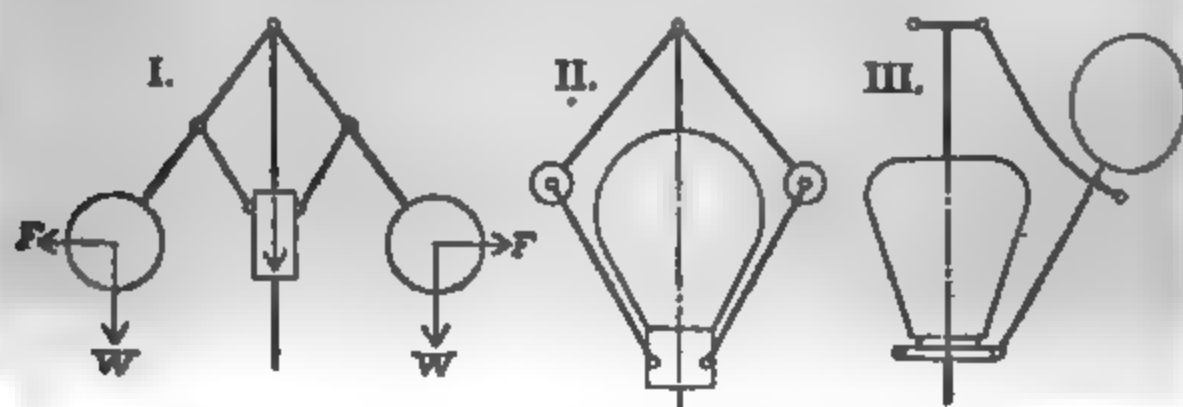


FIG. 525.—Types of the Fly-ball Governor: I. Common Low-speed Governor; II. High-speed Governor, Weight-loaded, III. The Proell Type.

(c) VARIOUS FORMS OF THE FLY-BALL GOVERNOR.—Three typical designs are outlined in Fig. 525, all weight-loaded, or with gravity as the force that acts against the centrifugal force of the balls. The first corresponds with Fig. 461 and is of the slow-running, self-balanced type, the weight on the central slide being small in comparison with that of the balls themselves. At II. is seen the high-speed governor, as in Fig. 489 and on Figs. 209, 212, 213; this governor is given a high rotary speed, and most of the counterforce is furnished by the central weight. Type III. is much used by German builders; the effect of thus placing the ball on another link of the mechanism will be discussed farther on. Sometimes fly-ball governors are loaded by means of springs instead of weights, or to supplement the weights, especially in the



Other types used on throttling engines: with full spring-loading the rotation-axis can be placed horizontal if more convenient.

f) FORCE-ACTION IN THE SHAFT-GOVERNOR.—The very simple force-action in the fly-ball governor is graphically represented on Fig. 525 I.: in steady running there is equilibrium; in adjustment the centrifugal force changes, and the difference between its actual value at any instant and that required to balance the weights is the net force which is available to move the governor. The analogous forces in the shaft-governor are shown on Fig. 526 I., the weight-arm AG is pivoted at A and rotates about the center O, and the spring-force  $S$  takes the place of weight. For this piece alone, the condition of equilibrium is

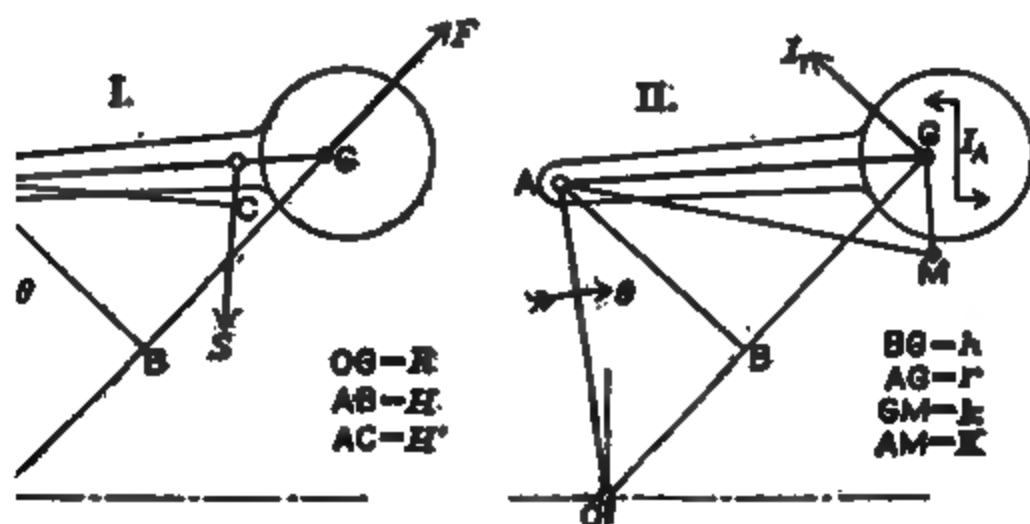


FIG. 526.—Force-action in the Shaft-governor.

$S \times AC$ . While the speed of the shaft is changing on account of changing load, the unbalanced part of the centrifugal force is supplemented by the inertia-action represented at II. When the shaft is receiving the angular acceleration  $\omega$ , the center of mass of the lever develops the linear acceleration  $OG \times \omega$ , and the mass  $M$  develops a force  $I = MR\omega$ , which acts with the lever-arm BG to move about A. At the same time the body AG has an angular acceleration (considered as about its center G), which is resisted by the moment of inertia  $Mk^2\omega$ ,  $k$  being, as usual, the principal radius of gyration. In this figure the weight-arm is so arranged that both inertia-effects act with the centrifugal force to produce the movement of the governor.

(e) **VARIOUS FORMS OF THE SHAFT-GOVERNOR.**—For the large number of governors of this type that have been brought out, the

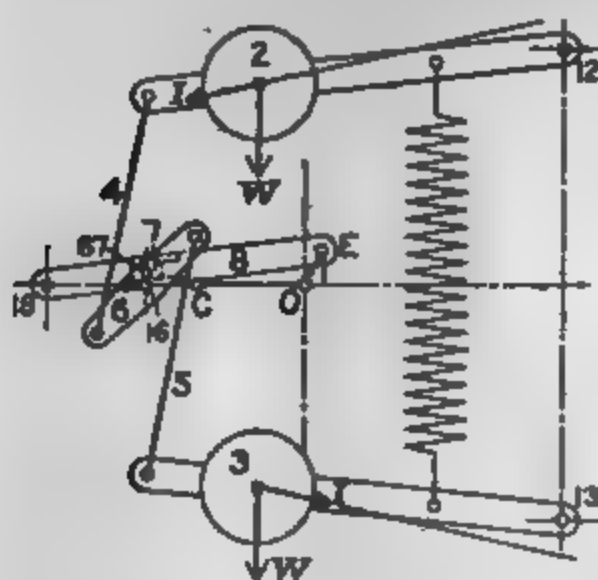


FIG. 527.—Diametral Symmetry.

best basis of classification is found in their arrangement with respect to the action of gravity and of inertia. The type showing the least departure from the fly-ball governor is represented by Fig. 527, where the weight-arms are pivoted on opposite sides of a diameter, and the main mechanism (which consists of the frame 1 and the moving pieces 2 to 6) is formed so as to give the arms 2 and 3 symmetrical movements. By

this arrangement weights and inertia-forces are completely balanced and neutralized, except as regards the comparatively insignificant actions on pieces 6 and 8; so that we have here a purely centrifugal governor. In four or five designs of this type, now mostly obsolete, the essential variety is found in the part of the mechanism that is directly concerned with moving the eccentric-pendulum; and this remark applies equally to the class that follows.

The larger class among the older designs is represented by Figs. 528 and 529. The essential feature is that the weight-arms are pivoted on a diameter and are connected in such a manner that their movements are nearly, if not quite, symmetrical with respect to the shaft-center. They are therefore balanced as to gravity, but the inertia-effects do not neutralize each other. The linear inertia-forces are drawn on both figures for the inner and the outer positions of the weights. In the Buckeye governor these force-lines pass so close to the pivots that the effective turning-moments are small, and these moments change from positive to negative as the weights swing out; in the Westinghouse design the positive effects are greater, but in both cases, as generally in related designs, the torque due to the angular inertia is negative.

are to be classed, then, as predominantly centrifugal governors, with a secondary inertia-action that more often hinders than helps the adjustment.

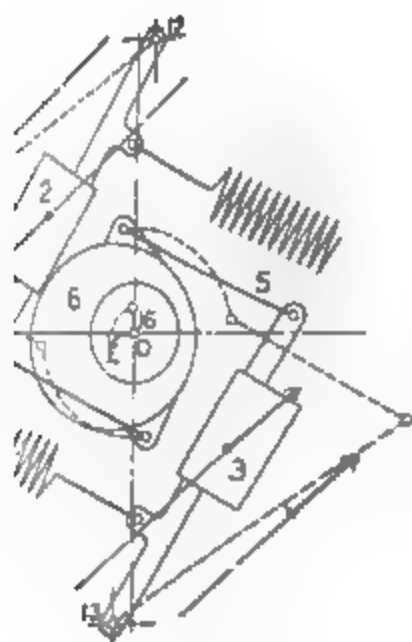


FIG. 528.—Buckeye Governor.

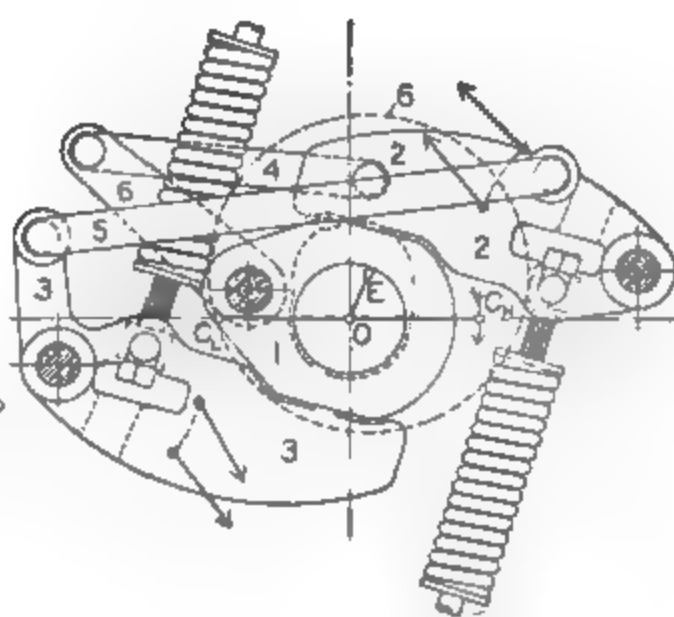


FIG. 529.—Westinghouse Governor.

Initially of the same type is the single-weight Sweet governor shown in Figs. 2 and 349, and skeletonized in Fig. 539. By giving sufficient weight on the eccentric-pendulum (see Fig. 2 y), this governor can be given almost a perfect gravity-

Its action is fully discussed in § 63 (c) to (f).

Now come now to the so-called inertia governors, which have to a considerable degree superseded the centrifugal type. These depend through centrifugal force, but depend very much upon proper and accurate adjustment and for steadiness. Contrasting examples are shown in Figs. 530 and 531. In the first, centrifugal force is as large, relatively speaking, as in purely centrifugal governors; but to it is added a powerful inertia-effect, which is wholly of the linear type. In the single-piece Rites governor centrifugal force-action is comparatively small, and linear inertia has a little effect because the center of mass  $G$  is so near to  $P$ ; but the moment of angular inertia is very large and has a powerful influence. The Rites governor is now used, of course, by probably a majority of the builders of steam engines, examples being given in Figs. 202, 205, and

207. A governor closely analogous in the predominant use of angular inertia is to be seen on Fig. 201, and another of similar general form is used on Figs. 206 and 215. Except on double-valve engines like Figs. 203, 216, and 217, the purely centrifugal shaft-governor has almost gone out of use. It will be noted that

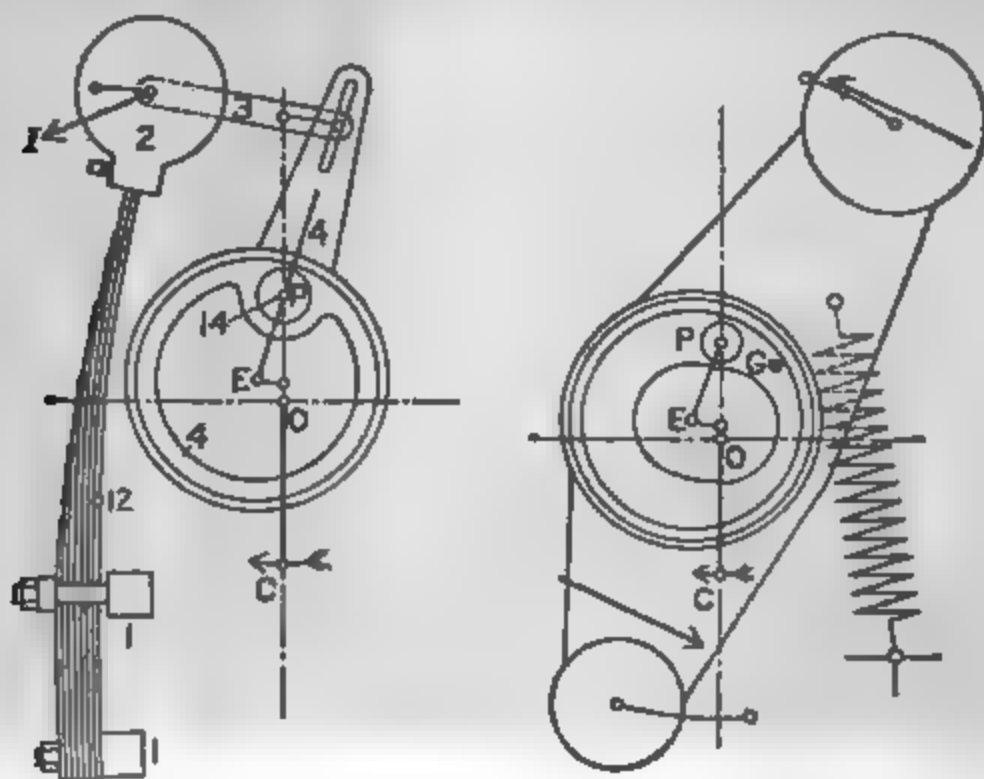


FIG. 530.—Robb-Armstrong Governor.

FIG. 531.—Rites Governor.

neither of the governors just shown is balanced as to weight—the gravity forces being relatively so small in effect that this is not found to be necessary.

### § 62. Special Principles and Methods.

As a preliminary to the analysis and discussion of the action of several representative governors, it will be necessary to set forth certain general principles of kinematics and of dynamics which find particular application in this line of work, and to develop from them methods of procedure, largely graphical, which will greatly facilitate the determinations to be made.

(a) REDUCTION OF FORCE FROM ONE PIECE TO ANOTHER.—In order to be able to transfer the force-action on one piece of a mechanism to another piece so as most easily to get a combined

or resultant effect, we shall make free use of the principle that *a force doing work at a certain rate upon one piece or at one point in a mechanism may be replaced by another force doing work at the same rate upon another piece or at another moving point*; the measure of work-rate being the product of the intensity of the force by its virtual velocity. By virtual velocity is meant the component of the total velocity (of the point of force-application) taken along the force-line, any other component being at right-angles to this line of action. Thus to reason by velocity-relations is simpler and more convenient, for present purposes, than to use the methods of graphic statics, and leads to identical results. It is therefore highly important to have an easy way of finding the relative velocity of any two moving pieces; and an extension of the idea of the instantaneous center meets this need in a most satisfactory manner.

(b) **RELATIVE INSTANTANEOUS CENTERS.**—In the discussion of the main engine-mechanism (refer to Fig. 112), we saw that the connecting-rod could be considered as turning for the instant about a point *P* which was located by the intersection of lines perpendicular to the paths of two points on the rod. In the fly-ball governor mechanism, Fig. 532, the analogous piece 3 has its rotation-center *P* found in the same way. Now this center is to be regarded, not so much as a point definitely located in space, as a point attached to the fixed piece 1; so that it is the common position of two points, attached to 1 and 3 respectively, which have the same velocity (here zero). Slightly broadening this conception—most naturally, perhaps, by considering that any “link” of a mechanism can be held fast as the fixed piece or “frame” without altering the relative movements of the links—we come to the following general definition:

*The instantaneous center between any two pieces of a mechanism is the position of common points (one attached to each piece, or definitely located on a plane fast to each piece) which have the same velocity, or which are moving together for the instant.*

**Theorem of Three Centers.**—The usual kinematic notation consists in numbering the several pieces or links and denoting the relative centers by the combined numbers. In Fig. 532, for

instance, we see that the joints B and C are to be designated as centers 23 and 34 (read "two-three", "three-four"). The problem now before us is, to find the common center between the non-adjacent moving links 2 and 4: for this we use the theorem of three centers, *That any three links of a mechanism must have their three relative instantaneous centers on a straight line.*

This fundamental idea is illustrated in Fig. 533, where we may consider links 1 and 3 as turning on link 2 at 12 and 23: then

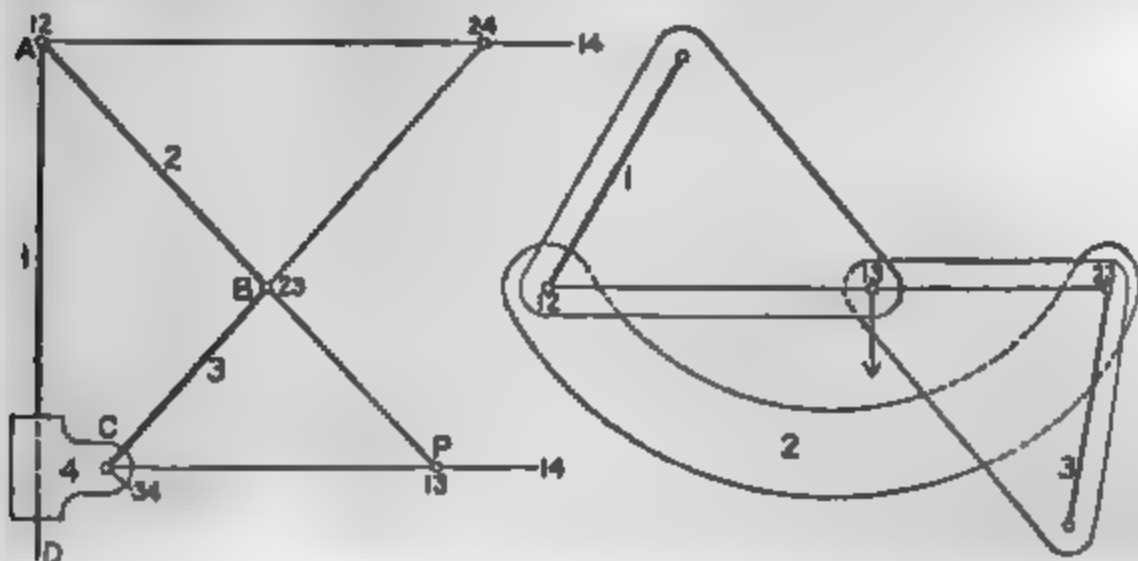


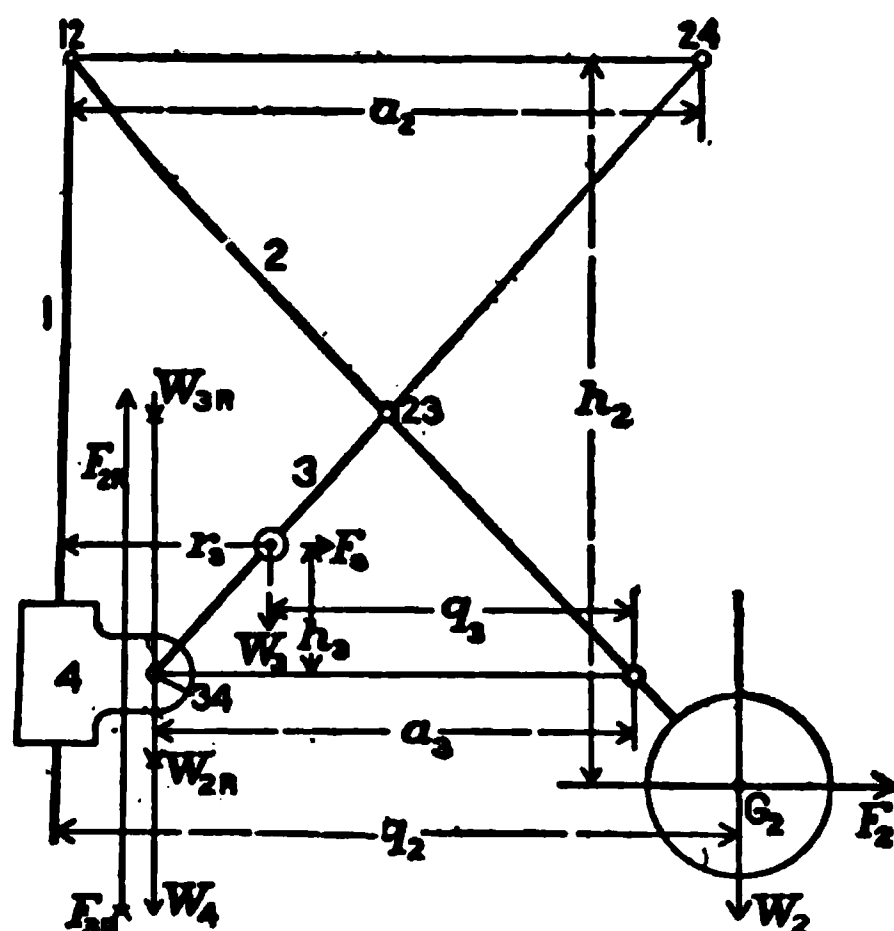
FIG. 532.—Instantaneous Centers in the Governor Mechanism.

FIG. 533.—The Theorem of Three Centers.

any point on 1 will have its velocity perpendicular to the radius from 12, any point on 3 to the radius from 23; and points with the same velocity can be got only by putting the two radii on the same line, which must be the line 12-23. In general, the center 13 can be anywhere on this line, the two links which it connects having opposite directions of rotation when 13 lies between 12 and 23, the same direction when 13 is beyond either of these primary centers.

Turning again to Fig. 532, and considering the three links 2, 3, and 4, we have that 24 must lie on the line 23-34; considering 1, 2, and 4, that it must lie on 12-14; wherefore the intersection of these two lines determines the actual center. Since piece 4 slides on a straight line (an arc of infinite radius), its center 14 is at an infinite distance from the line AD, and any perpendicular from AD is a radius to 14. Note further that the center 13 is at the intersection of 12-23 and 34-14.

RELATIVE VELOCITY AND REDUCTION OF FORCE.—To see how the preceding principles are applied, consider first Fig. 534;



—Reducing all the Forces to the Slide, in a Fly-ball Governor.

A general proportions is taken directly from the governor. On piece 2 act the centrifugal force  $F_2$ , and the gravity on piece 3 act  $F_3$  and  $W_3$ . It is most convenient to reduce the force-actions to the slide 4, partly because that is a very simple movement, chiefly because this movement is utilized for the control of the cut-off. With the governor turning about the center 12, link 2 is subjected to the moments

$$+F_2 h_2, \quad -W_2 q_2.$$

A point of common velocity determined at 24, and with the act that in this case all points on 4 have the same movement only replace  $F_2$  or  $W_2$  at  $G_2$  by an equivalent force perpendicular to the lever-arm  $a_2$ , to get the reduced effect force can then be transferred to any desired combination 34. These reduced forces in Fig. 534 have the values

$$F_{2R} = F_2 \frac{h_2}{a_2}, \quad W_{2R} = W_2 \frac{q_2}{a_2}; \quad \dots \quad (331)$$

and are laid off at 34, in combination with the similar reduced forces from 3,

$$F_{2R} = F_2 \frac{h_2}{a_2}, \quad W_{4R} = W_4 \frac{q_4}{a_4}$$

The resultant of the forces from pieces 2 and 3 must balance the slide-weight  $W_4$ .

It appears therefore that the dimensions to be taken from the layout of a governor of this type for any link  $n$  are as follows:

$r_n$  = radius from the rotation-axis, which is a factor in the centrifugal force;

$h_n$  = lever-arm of the centrifugal force, measuring its tendency to turn the piece about the center  $1n$ ;

$q_n$  = lever-arm of weight about center  $1n$ ;

$a_n$  = radius to which the forces are to be reduced in preparation for their transfer to the slide.

In the rather more general case, represented by Fig. 535, where reduction is to be made to the point E on a link 4 that turns, the procedure for any other link 2 is as follows:

Find the common center 24 and measure the radii  $12-24=a_2$  and  $14-24=a_4$ ; then if  $\theta_2$  and  $\theta_4$  be simultaneous angular velocities

of links 2 and 4 about their centers 12 and 14, the common linear velocity at 24 is  $v_{24} = a_2 \theta_2 = a_4 \theta_4$ ; whence

$$\frac{\theta_2}{\theta_4} = \frac{a_4}{a_2} = \frac{14-24}{12-24} \quad (332)$$

Now the virtual velocity of  $F_2$  is  $v_2 = \theta_2 h_2$ , and that of  $F_{2R}$  is  $v_E = \theta_4 h_4$ ; wherefore

$$F_2 \theta_2 h_2 = F_{2R} \theta_4 h_4$$

$$F_{2R} = F_2 \frac{\theta_2 h_2}{\theta_4 h_4} \quad (333)$$

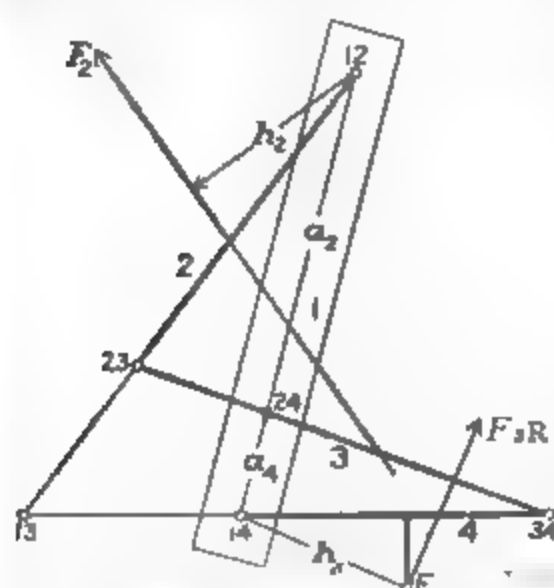


FIG. 535. Force-reduction to a Link that turns.

Another way of finding relative velocities, less general in its





where  $k$  is the radius of gyration and  $\theta$  must have the same value as in 534 when simultaneous component-motions are assumed. Then for the total energy we have

$$E = 1/2 M \theta^2 (R^2 + k^2). \quad \dots \dots \dots$$

Fig. 536 suggests that by simply drawing  $k$  perpendicular to  $R$ , we get at  $M_c$  a point where the whole mass may be concentrated.

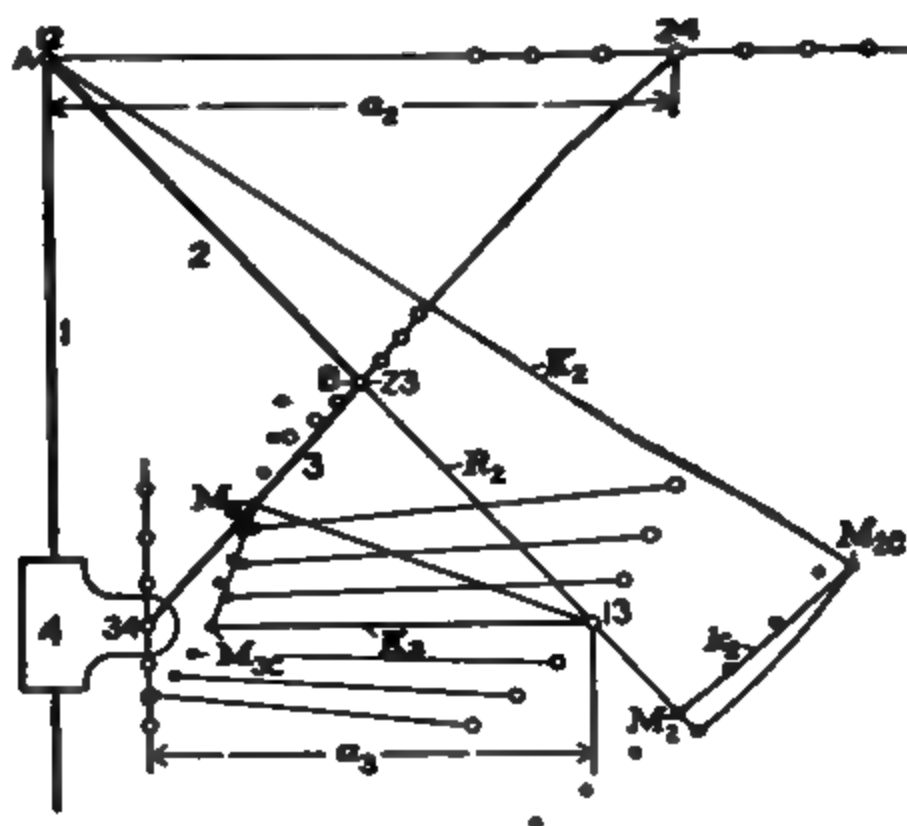


FIG. 536.—Mass-relations in the Fly-ball Governor.

for simple linear velocity due to turning about the instantaneous center, because  $K^2 = R^2 + k^2$ .

Now for the simple governor of Figs. 534 and 536, where, in effect, the mass must merely be transferred to another point on its own piece, we have the relations

$$M_1 K_1^2 = M_{1R} a_1^2; \quad M_2 K_2^2 = M_{2R} a_2^2;$$

$$\text{or} \quad M_{1R} = M_1 \left( \frac{K_1}{a_1} \right)^2; \quad M_{2R} = M_2 \left( \frac{K_2}{a_2} \right)^2. \quad \dots \dots (337)$$

the more general case illustrated in Fig. 535,

$$M_2 K_2^2 \theta_2^2 = M_{2R} h_4^2 \theta_4^2,$$

$$M_{2R} = M_2 \left( \frac{K_2}{h_4} \right)^2 \left( \frac{\theta_2}{\theta_4} \right)^2 \dots \dots \dots (338)$$

### § 63. The Problem of Regulation.

5) FORCE-ACTIONS IN A FLY-BALL GOVERNOR.—We shall now examine the behavior in steady running of the fly-ball governor at Fig. 461, which is reproduced in Fig. 537 and has already diagrammed in Figs. 534 and 536. The weights of the several parts are:

$W_2$  = total weight of arm 2 = 56.87 lbs.;

$W_3$  = weight of link 3 = 0.97 lb.;

$W_4$  = total effective weight on slide 4 = 38.0 to 66.6 lbs., including all suspended parts, and varying with the location of the counterpoise on its arm (see Fig. 461). For one side of the governor, as in Fig. 537, we use half of this weight.

The movement of the weight-arm 2 is divided into four equal parts from position 2 to 6, and the determination is carried one beyond this range at each end. The measurements taken from the diagram of the governor, as defined on page 376, are given in Table 63 A, all in inches. Of the two mass-points marked by concentric circles on each piece, the inner is the true center of mass and the outer is the center for the effective centrifugal force. A knowledge of the fact that the true effect of centrifugal force, in a fly-ball governor, cannot be got by taking this force as at the center of mass or of gravity, together with a suggestion of methods for finding the proper point of mass-concentration, will be found in Appendix to this chapter, page 435. Here the weight of the balls acts at the inner point, but the centrifugal force is to be taken as if the whole mass were at the outer point, with an

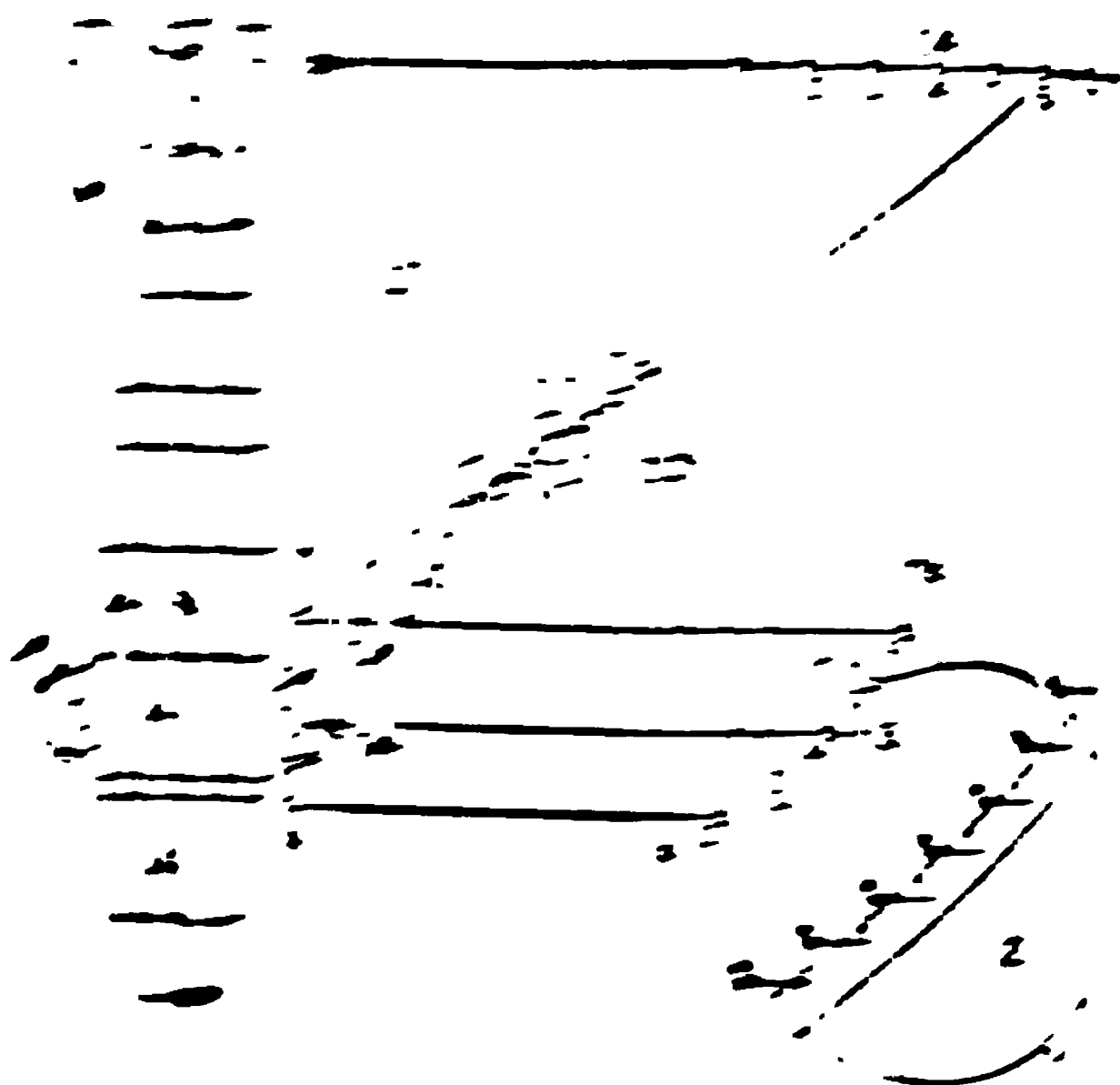


Fig. 1. - Diagram of the belt and pulley system.

Table 1. - Data for the belt and pulley system.

No.	Pulley 1				Pulley 2			
	1	2	3	4	5	6	7	8
1	9.76	15.75	8.75	2.20	3.20	3.20	5.45	4.4
2	10.74	15.75	11.32	3.45	3.45	3.45	5.46	7.1
3	11.72	16.22	11.34	13.75	4.14	2.38	4.21	8.1
4	12.68	12.50	12.45	11.23	4.31	2.75	6.71	6.5
5	14.58	12.62	12.31	13.11	4.52	2.52	7.13	9.4
6	14.44	11.64	14.11	14.20	4.74	2.38	7.48	10.7
7	16.00	10.64	14.46	15.50	4.25	1.28	7.48	10.4

For convenience in the final determination of the balance speed, we at first calculate the centrifugal forces for a speed of

.P.M. of the governor, and reduce them to slide 4. The  
l formula, for radius  $r$  in inches—see § 35 (b)—is

$$F=\frac{W}{g}\theta^2\frac{r}{12}; \quad \theta=2\pi\frac{N}{60}=0.10472\ N; \quad . \quad . \quad . \quad (339)$$

gular velocity  $\theta$  being expressed in radians per second.  
king  $N=100$  and reducing, we get the general relation

$$F=0.28416\ Wr; \quad . \quad . \quad . \quad . \quad . \quad (340)$$

h the weights given above we have

$$F_2=16.16\ r_2, \quad F_3=0.276\ r_3. \quad . \quad . \quad . \quad . \quad (341)$$

ese formulas, with (331), we calculate the values in Table  
l expressed in pounds or in pound-inches It may be  
d that the slide-rule is very effective and convenient for  
of work, and that nearly all of the calculations given in  
ter are of the degree of accuracy secured by that method.

ABLE 63 B. REDUCTION OF CENTRIFUGAL FORCE.

Arm 2.			Link 3.			Total.
$F_2$	$F_2h_2$	$F_{2R}$	$F_3$	$F_3h_3$	$F_{3R}$	$F_R$
56.8	2468	290.7	0.97	3.18	0.49	291.2
53.7	2620	272.0	1.04	3.26	0.45	272.5
49.7	2719	252.2	1.11	3.30	0.41	252.6
44.5	2760	231.6	1.19	3.26	0.37	232.0
38.8	2761	210.8	1.25	3.15	0.33	211.1
31.9	2706	189.5	1.31	2.96	0.30	189.8
24.0	2602	168.0	1.36	2.70	2.26	168.3

1

2

3

4

5

6

7

w the moment  $F_2h_2$  varies; involvin? the product  $r_2h_2$ .  
t when the arm is at 45°, and decreases in either direction  
osition;  $F_3h_3$  varies in the same general manner, but  
o light and so near the axis of rotation that its force-  
gni-

A similar set of values for the weight-forces is given in Table 63 C, where each of the last three columns is the summation of columns 2 and 4 with the value of  $W_4$  at its own head, and Cases A, B, and C represent the weight-action for the adjustable poise  $W$  (see Fig. 461) at its inner, middle, and outer positions respectively.

TABLE 63 C. REDUCTION OF WEIGHTS TO SLIDE 4.

Position.	Arm 2. $W_2 = 56.87$		Link 3. $W_3 = 0.97$		Total Weight $W_R$ on Slide 4		
	$W_2 g_2$	$W_R$	$W_3 g_3$	$W_{gr}$	Case A $W_4 = 19.0$	Case B. $W_4 = 26.15$	Case C $W_4 = 33.3$
1	544.0	64.05	4.90	0.76	83.81	90.96	98.11
2	601.5	62.50	5.49	0.75	82.25	89.40	96.55
3	656.5	61.00	6.02	0.74	80.74	87.89	95.04
4	708.2	59.32	6.92	0.73	79.05	86.20	93.35
5	757.5	57.80	7.24	0.73	77.53	84.68	91.83
6	803.0	56.20	7.22	0.72	75.92	83.07	90.22
7	845.0	54.55	7.16	0.72	74.27	81.42	88.57
	1	2	3	4	5	6	7

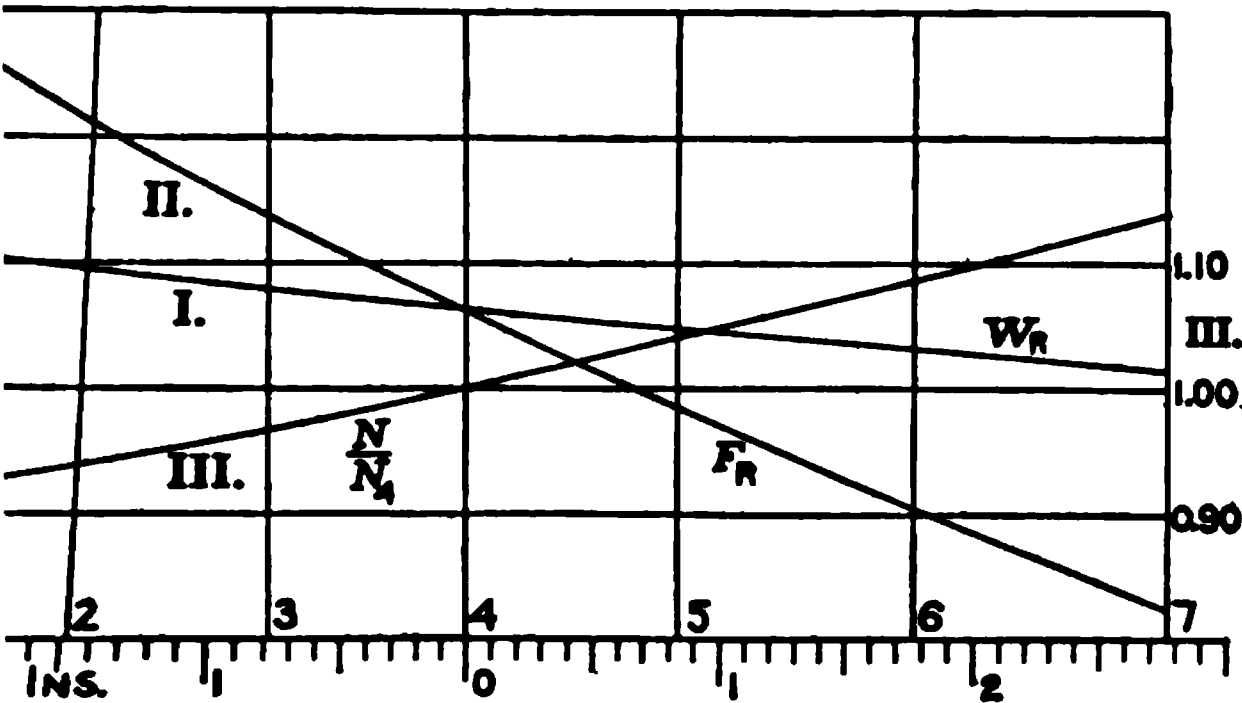
(b) REGULATION BY THE GOVERNOR IN FIG. 537.—Now the speed of rotation  $N$  of the governor must have such a value, for each position and in each case, that  $F_R$ , given for 100 R.P.M. in column 7 of Table 63 B, will be made equal to  $W_R$  as given in column 5, 6, or 7 of Table 63 C. Thus for position 4, Case B,  $F_R$  must have the value 86.20 instead of 232.0, or must be 0.3715 as great as for 100 R.P.M. Since  $F$  varies as  $N^2$ , we take the square root of this ratio and get the actual speed to be 60.9 R.P.M. What the corresponding engine-speed will be depends upon the relative sizes of the pulleys in the belt drive; which are here 15" and 20", so that the engine turns four-thirds as fast as the governor.

The regulating-action of an engine-governor is to be judged chiefly by the variation in its steady-running speed, as here set forth in columns 1, 4, and 7 of Table 63 D. The same thing is expressed in slightly different terms in columns 2, 5, and 8, where each speed is compared with that at position 4 as unity. This governor is by no means a close regulator, the total variation from position 2 (resting on stop-ring) to position 6 (highest permitted

BLE 63 D. REGULATION BY GOVERNOR IN FIGS. 461, 537.

n.	Case A.			Case B.			Case C.		
	$N$	$N/N_4$	$F_{R1}$	$N$	$N/N_4$	$F_{R1}$	$N$	$N/N_4$	$F_{R1}$
	53.6	0.919	99.4	55.8	0.917	108.0	58.0	0.916	118.0
	54.8	0.941	93.1	57.3	0.939	101.1	59.3	0.937	109.5
	56.5	0.969	86.2	58.9	0.968	93.7	61.3	0.967	101.5
	58.3	1.000	79.1	60.9	1.000	86.2	63.4	1.000	93.4
	60.6	1.039	72.1	63.3	1.040	78.3	65.9	1.040	84.8
	63.2	1.083	64.8	66.1	1.085	70.5	68.8	1.087	76.3
	66.4	1.138	57.5	69.5	1.141	62.4	72.5	1.145	67.6
	1	2	3	4	5	6	7	8	9

) being 14.2, 14.6, and 15.0 per cent. of the mean speed at three cases. Note that moving the poise merely changes absolute speed, with only a minute influence upon its manner of regulation. The last column in each group (3, 6, 9) shows the  $F_R$  would have if the speed were kept constant at the



538.—Curves of Regulation for Case B, Tables 63 C, 63 D.

esponding to position 4, each of these columns being multiplying all the values in column 7 of Table 63 B by ratio—which is 0.3715 for Case B, as shown above. By of these forces with the resultant weight-effects in primary measure of the stability of the





ernor-gear for which determinations have been made. The curves for Case B, and are plotted, I. from col. 6, Table 63 C; II. in col. 6, 63 D; and III. from col. 5, 63 D. Curve III. shows the variation in speed that will make II. agree with I. The divergence between I. and II. shows the force which, for a given speed, acts to hold the governor in a certain position. The less this divergence, the more sensitive and unstable the governor is. We shall presently consider the simple problem of making the balancing-speed more nearly constant: but how near the governor may be brought to isochronous running, with due regard to stability, is a question that can be answered only after a study of its action in adjustment—and then, perhaps, not very closely.

(c) ANALYSIS OF A CENTRIFUGAL SHAFT-GOVERNOR. — The governor drawn in skeleton-outline at Fig. 539 is like that on the engine in Fig. 2, all the dimensions and data being taken from an actual design. Piece 2 is the weight-arm, with its center of gravity  $G_2$ , a little off the center-line because the head is made hollow and partly filled with shot, which lie as far out as possible when the engine is running. Piece 4 is the eccentric-pendulum, with enough mass, central at  $G_4$ , to produce the gravity-balance described in § 61 (e). The strap 5 is fastened at the upper end to the arc on 2, at the lower end is gripped in a clamp jointed to the spring-end. The forces acting on 2, 3, and 4 are to be accurately determined and reduced to the point E on 4. As to the spring and strap, it will be assumed that they are closely enough represented by a mass concentrated at the joint 56, and equal to the strap-clamp plus a reduced mass for the spring. The latter is determined by integrating an expression which takes account of the fact that the cross-section varies almost uniformly from the inner toward the outer end, and that the deflection of any point on the spring varies as the cube of the distance from the fixed end—this furnishing a ratio of the virtual velocities. Then the primary data as to the links of the mechanism are as follows:

Piece	2	3	4	56	lbs.
Weight	32.9	5.0	24.5	4.5	lbs.
vr. $k$	5.75	4.86	4.08	0.0	ins.

Certain important dimensions that are to be used in the calculation are given on the figure, and those essential to the valve-action are shown at II.

(d) KINEMATIC RELATIONS.—The centers 13 and 24 being inconveniently distant (especially 13), we do not follow the method of Eq. (332), but instead use a construction which can be made within a smaller space. This is outlined at III., for position 5, and is carried through as follows:

The velocities of points B and D, due to turning about A and P, will be perpendicular to the radii AB and PD. If they are resolved into rectangular components, along and across the connecting-link BD, the components BM and DN must be equal: so that, having BH, we could transfer BM to DN and from it get DK. Then if  $\theta_2$  and  $\theta_4$  are the angular velocities about A and P, we have

$$BH = \theta_2 \times AB, \quad DK = \theta_4 \times PD;$$

whence

$$\frac{\theta_2}{\theta_4} = \frac{BH}{DK} \frac{PD}{AB}.$$

To get the angular velocity of link 3, we note that the difference between the cross-components MH and NK is due to  $\theta_3$  (no center specified); so that  $(MH - NK) = \theta_3 \times BD$ . The construction in its most compact and serviceable form is shown at Fig. 540 I., where bd, of any convenient length, represents the velocity along link 3, dh is perpendicular to it, and bh and bk are the total velocities of the points B and D, made perpendicular to AB and PD. Then the operations actually carried out are

$$\frac{\theta_2}{\theta_4} = \frac{bh}{bk} \frac{PD}{AB}, \quad \frac{\theta_3}{\theta_4} = \frac{hk}{bk} \frac{PD}{BD} \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (342)$$

A little consideration will show that the figure bhk is a miniature of that which would be got by drawing radii from the center 13 to the points B and D.

For the forces which act through  $G_3$ , the relative virtual velocities are to be found directly. In Fig. 539 III., the line OG is a radius

0, or is the line of centrifugal force; and GF, perpendicular is the direction of linear inertia. Resolving the total velocity of the center G into components GR and RL, we get the virtualities  $V_p$  and  $V_i$ ; and these are to be compared with the

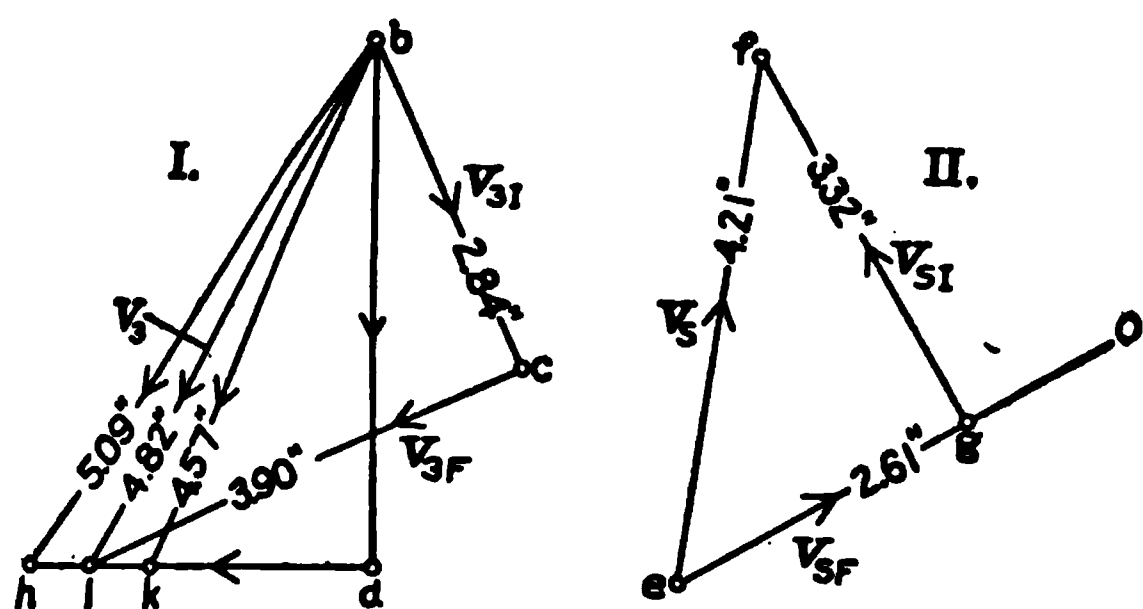


Fig. 540.—Special Constructions for Velocity.

of E, which is to that of D in the ratio of PE to PD. In I., l is midway between h and k (note that QL is half way HM and KN), and lc and cb have the directions OG, GF. In the reduction of mass we shall need also the total velocity of the mass at 56, then the calculations to be made are

$$V_s = V_E \frac{bl}{bk} \frac{PD}{PE}; \quad V_{sF} = V_s \frac{cl}{bl}; \quad V_{sI} = V_s \frac{bc}{bl}. \quad (343)$$

For the spring-force, acting along the strap 5, and for the forces on the mass at 56, a similar determination of relative velocities is made. In Fig. 540 II. the total velocity  $V_s$  along the strap 5 is resolved into components in the directions of centrifugal force and linear inertia of the mass at 56, again for position 5. Measuring the perpendicular distance (12-S) or AS (from A to the strap), we get the relations

$$V_s = V_E \frac{AS}{PE} \frac{\theta_2}{\theta_4}; \quad V_{sF} = V_s \frac{eg}{ef}; \quad V_{sI} = V_s \frac{gf}{ef}. \quad (344)$$

From these various equations we get the numerical results given in Table E, all the linear velocities being in terms of  $V_E$  as unity.

TABLE 63 E. VELOCITY-RELATIONS IN THE AMES GOVERNOR.

Position	1	2	3	4	5	
$\theta_2/\theta_1$	1.444	1.452	1.475	1.523	1.606	1
$\dot{\theta}_2/\dot{\theta}_1$	-0.000	-0.050	-0.103	-0.167	-0.261	2
$V_1$	1.924	1.929	1.942	1.974	2.030	3
$V_2$	1.384	1.428	1.485	1.556	1.642	4
$V_3$	1.343	1.294	1.254	1.216	1.196	5
12 B	6.13"	5.78"	5.44"	5.11"	4.78'	6
$V_6$	1.311	1.243	1.190	1.153	1.138	7
$V_{ar}$	0.967	0.887	0.816	0.754	0.705	8
$V_{su}$	0.897	0.875	0.871	0.873	0.898	9

(e) REACTION OF THE VALVE-GEAR.—Before proceeding to calculate and balance the centrifugal and spring forces, we must determine the effect of the reaction of the valve-gear upon the eccentric. In a Corliss engine only a very small force is exerted by the valve-gear upon the governor—merely the pressure of the trip-arm upon the cam while in the act of unhooking, due to friction of the latch-edges under the pull of the dash-pot. But with a shaft-governor the whole force required to move the valve-gear reacts upon the eccentric: this reaction varies widely in effective magnitude throughout the revolution, and has an average resultant which helps to determine the balancing-speed.

The problem before us is essentially the same as that of the crank-pin pressures, discussed in § 38 (a) to (d); but in this case the simpler methods give amply accurate results. It is better to separate inertia and valve-resistance, partly because results can be reached more easily, chiefly because one is a direct function of the speed and the other is not. In this engine the total mass of the valve-gear, reduced to the outer end of the rocker-arm (to the wrist-pin), and including the whole of the eccentric-rod, is equal to a weight of 38.7 lbs. In general, we should concentrate at the eccentric-center the mass of the eccentric-strap plus a part of the rod, whereupon this mass will have simple centrifugal force—the rest of the rod is to be added to the sliding parts. With the small eccentric-pin (see Fig. 2), we here put half the rod, or 8.5 lb

the eccentric, the remaining 30.2 lbs. at the wrist-pin on the rocker.

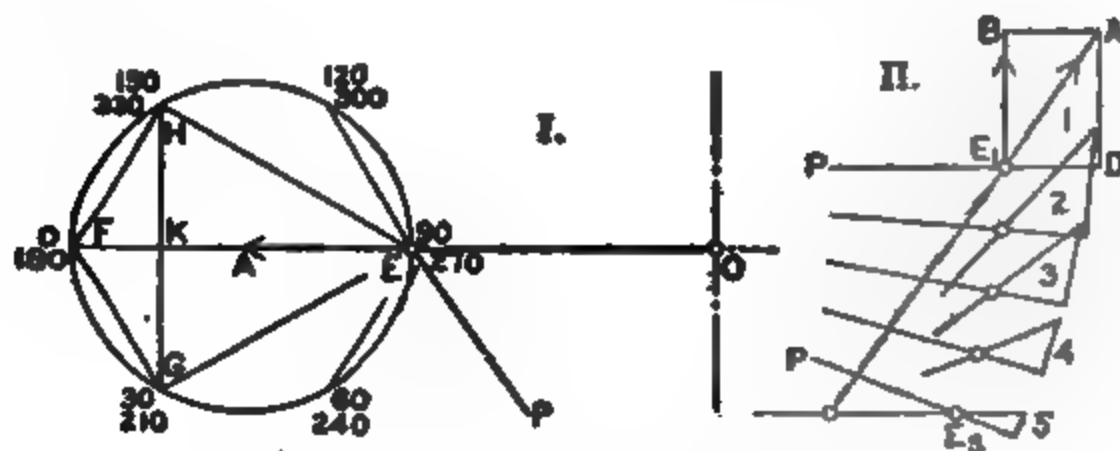


FIG. 541.—Inertia of the Valve-gear.

**Valve-gear Inertia.**—The force-action upon the eccentric, due to the reciprocating mass of the valve-gear, is illustrated in Fig. 541. It is a simple matter to calculate the ideal centrifugal force  $F_0$ —see § 35 (a)—and then use the formula for harmonic motion,  $F = F_0 \cos \alpha$ . This force  $F$ , always acting along the stroke-line, is laid off from the center  $E$  in Fig. 541 I., after the manner of Fig. 159. In effect, we imagine the eccentric-radius  $OE$  to stand still and the stroke-line to rotate backward around  $E$ . The polar diagram of the inertia  $F$  is a circle like the Zeuner valve-circle, over which the rotating radius sweeps twice during one revolution, as indicated by the angle-numbers.

Consider now any pair of forces as  $EG$  and  $EH$ , symmetrically placed with reference to the dead-center line  $EF$ : the opposing components  $KG$  and  $KH$  neutralize each other as to any tendency to produce resultant movement of the eccentric-center, but the two components  $EK$  tend to move  $E$  radially outward. To find the average effect—the angle-base being equivalent to a time-base—we must average up  $EK = F \cos \alpha = F_0 \cos^2 \alpha$  for half a revolution, or integrate

$$\int_{-\frac{\pi}{2}}^{+\frac{\pi}{2}} F_0 \cos^2 \alpha \, d\alpha = F_0 \left[ \frac{1}{2} \sin 2\alpha + \frac{1}{2} \alpha \right]_{-\frac{\pi}{2}}^{+\frac{\pi}{2}} = \frac{1}{2} \pi F_0. \quad (345)$$

Product of force by angle or time, we take out the

factor  $\pi$  and get  $\frac{1}{2}P_r$  as the mean radial force. This force is off at EA in I.; and in II. the EA for each governor-position resolved into rectangular components, ED along PE, EB tend to move E outward along its path, acting with the spring against the positive centrifugal force of the weight-arm.

In calculating the values given in Table 63 G, line 5, we can take half of the centrifugal force due to the slide-mass of 30.2 lbs. and add it to the whole of that due to the 8.5 lbs. at the eccentricity, or, to reduce the number of operations, can make the determination of Fig. 541 for one-half of  $(30.2 + 2 \times 8.5)$  or for 23.6 lbs.

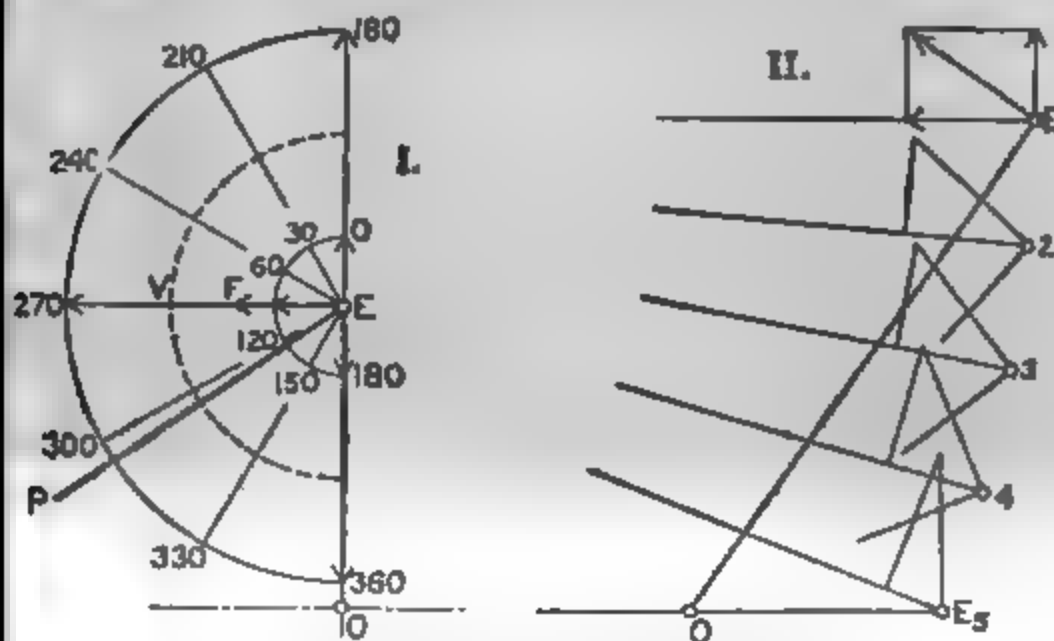


FIG. 542.—Effect of Valve-resistance.

**Valve-resistance**—The resistance of the valve is one of the most uncertain and indeterminable forces in the engine. It seems reasonable to assume that this will be an approximately constant force, acting against the movement in each stroke. Some experiments that have been made indicate a value of about 50 lbs., applied to the outer end of the rocker, for the engine under consideration. This value is used in Fig. 542 I., modified by the pressure of the steam on the end of the valve-rod, which changes to 20 lbs. on the forward stroke, 80 lbs. on the return stroke of the valve. Laying off this valve-force from E, as in Fig. 541, we get the action represented by the two full-line semicircles, and the dotted curve for the average of both strokes. Here the

force-action is symmetrical with respect to a line at right angles to the eccentric-arm; resolution and integration give the mean force  $EF = 2/\pi \times EV$ , or  $EF = 0.6366 \times 50 = 31.83$  lbs. In II. this force is laid off from E, perpendicular to each eccentric-radius; and the components perpendicular to PE are added to the spring-force in Table 63 I.

It will be noted that we have here, especially in Fig. 542, disturbing force-actions of the first magnitude. According to the diagram at I. there is one position, between  $180^\circ$  and  $210^\circ$  of the eccentric-travel, where the full force of 80 lbs., plus the inertia shown in Fig. 541 I., is tangential to the path of E. It is by making the time of action short that the momentum given to the governor-mass is kept within practically negligible limits—which is a strong reason why the shaft-governor is not well adapted to slow-moving engines. Only in a closer study of the subject than is here permissible would an application be made of the methods used for the fly-wheel discussion in § 36 (d), with a view to finding how great is the small periodic oscillation due to this variable force. It is evident that the valve-gear reaction completely overshadows any force-variation due to the periodic fluctuation in the speed of the shaft, even when we consider the heavy inertia-governor in Fig. 531.

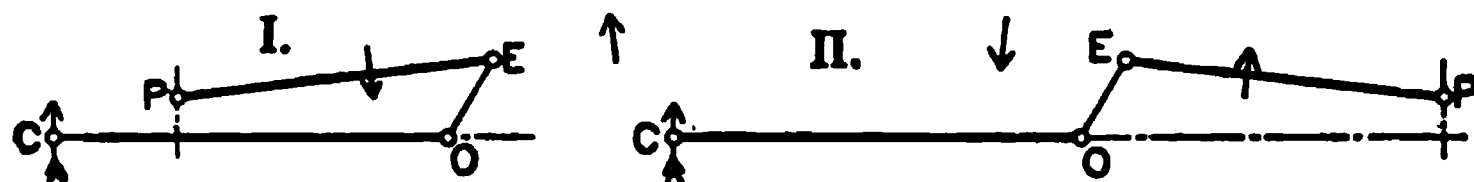


FIG. 543.—Effect of Eccentric-friction.

*Eccentric-friction.*—Friction in the various joints of the governor-gear tends simply to hinder its movement, with an influence upon its action in adjustment: friction in the joints of the valve-gear enters as a minor component into the resistance just considered: but friction of the strap upon the eccentric—which may rise to a considerable magnitude under certain conditions—takes the form of a nearly constant torque, tending to turn the eccentric backward, and to rotate the eccentric-pendulum upon the pivot P. In the case under consideration, with only a small pin, this action is not important. In general, the effect of excessive friction on the

eccentric will be to make a certain nearly uniform change in the running-speed, without affecting the variation of the speed in regulation. Whether this change will be an increase or decrease depends upon the arrangement of the eccentric-pendulum with reference to the crank, as illustrated in Fig. 543. In the first case, the friction-torque tends to throw E out, helping the spring, and requiring an increment of speed to balance it; in the second case, opposite conditions prevail. It is of interest to note that if the eccentric-strap gets hot and sticks fast, an engine arranged as in Fig. 543 II. will simply be shut down, while one with the other arrangement is likely to "run away" and be seriously damaged.

TABLE 63 F. MEASUREMENTS FOR CENTRIFUGAL FORCE, FROM FIG. 539.

Position.	1	2	3	4	5	
Piece 2 { $R_2$	16.23	16.97	17.66	18.30	18.90	1
{ $H_2$	5.63	5.23	4.78	4.28	3.73	2
Piece 3 { $R_3$	11.48	10.59	9.66	8.69	7.65	3
Piece 4 { $R_4$	2.52	2.02	1.52	1.02	0.54	4
{ $H_4$	5.26	5.30	5.31	5.23	4.73	5
Piece 56 { $R_{56}$	11.28	10.74	10.23	9.75	9.30	6
Valve-gear r	3.06	2.54	2.04	1.59	1.26	7

(f) BALANCING FORCES IN FIG. 539 — Here again, as under 537, we at first calculate the centrifugal forces for 100 R.P.M. Substituting in Eq. (340) the weights given in Arts. (c) and (e), we have in the formula  $F=CR$  the following values for the coefficient  $C$ :

Piece	2	3	4	56	V.G.
Weight	32.9	5.0	24.5	4.5	23.6
$C$	9.35	1.42	6.96	1.28	6.71

Radii from the center O to each center of gravity are given in Table 63 F, measured in inches and ready for substitution in (340). From these are calculated the centrifugal forces in Table 63 G, which are given, not because they are to be used directly, but rather for the sake of getting some idea of their relative magnitude.



TABLE 63 G. CENTRIFUGAL FORCES IN FIG. 539, AT 100 R.P.M.

Piece.	1	2	3	4	5	
2	151.8	158.6	165.0	171.0	176.6	1
3	16.3	15.0	13.7	12.3	10.9	2
4	17.7	14.0	10.6	7.1	3.8	3
56	14.4	13.8	13.1	12.5	11.9	4
V.G.	20.5	17.0	13.7	10.7	8.4	5

For pieces 2 and 4 we use the method of Eq. (333), § 62 (c). The lever-arm  $H$ , equal to AB on Fig. 526 or AF on Fig. 539, is given in Table 63 F; for  $h_4$  we use PE or Q, equal to 6.75". The angular velocity-ratios are given in Table 63 E, as are also the ratios of virtual velocities by which we must multiply the actual centrifugal forces of 3 and 56 in order to get their reduced forces at E, according to the relations

$$F_{3R}=V_{3F}F_3, \quad F_{56R}=V_{56F}F_{56} \quad . \quad . \quad . \quad . \quad . \quad (346)$$

The results of these calculations are given in Table 63 H, where the effective component of the valve-gear inertia, line 5, is taken from Fig. 541.

TABLE 63 H. REDUCED CENTRIFUGAL FORCES.

Piece.	1	2	3	4	5	
2	+183.0	+178.4	+172.3	+165.3	+156.8	1
3	— 22.5	— 21.5	— 20.4	— 19.2	— 17.8	2
4	— 13.8	— 11.0	— 8.3	— 5.5	— 2.7	3
56	— 13.9	— 12.2	— 10.7	— 9.4	— 8.4	4
V.G.	— 17.1	— 13.7	— 10.2	— 67.	— 3.2	5
$\Sigma F_R=F_{100}$	+115.7	+120.0	+122.7	+124.5	+124.7	6

Table 63 I contains the final set of values in this computation. The increment of spring-deflection is  $\Delta f$ , in inches, and the scale of the spring is 92.0 lbs. to 1 in. of deflection. Then the increments of spring-force have the values  $\Delta S$ , and with the initial tension 718 lbs. (experim<sup>er</sup>+<sup>a</sup>l) we get the forces  $S$  as acting along the strap. These by the virtual velocity-ratio  $V_s$  from

Table 63 E, we have the reduced spring-forces  $S_R$ . To these are added the effective component of the mean valve-resistance from Fig. 542, and the result is the total balancing force  $B$ . Dividing this  $B$  by  $F_{100}$ , and multiplying 100 by the square-root of the ratio, we get the balancing-speed  $N$ . For this last calculation the slide-rule is not effective, and five-place logarithms must be used.

TABLE 63 I. BALANCING FORCES AND SPEEDS.

Position.	1	2	3	4	5	
$A/$	0 00	0 83	0 78	0 74	0 70	1
$AS$	00 0	76 4	71 8	68 1	64 4	2
$S$	718 0	794 4	866 2	934.3	998 7	3
$S_R$	941.3	987 4	1030 6	1077 3	1136 5	4
V.G.	18 4	19 6	21 8	25 2	29 8	5
$B$	959 7	1007 0	1052.4	1102 5	1166 3	6
$N$	288 0	289 7	292 9	297.6	305 8	7

It will be noted that the regulation shows quite a variation in character, being close at first, coarser as the eccentric moves in. Referring to Fig. 552, where the curve of M.E.P. is given, we see that this manner of variation tends to make the regulation uniform if referred to the load on the engine rather than to the position of the eccentric. A further inspection of this curve shows that the regulation is closer than at first sight appears; for whereas the total variation of  $N$  in the table is about 6 per cent. over the whole range of the governor, it appears that the effective range of loading is, roughly, from position 2 to position 4.1, with a speed-change of only about 4 per cent.

(g) CONTROL OF THE REGULATION.—The methods developed and applied in the preceding discussion can be used in the solution of any problem as to regulation that may arise: and before passing to the question of adjustment and stability, we shall consider briefly how the regulation can be varied in character in the two types of governors.

In Fig. 544, the primary element of the fly-ball governor is shown at I. in its simplest form, or as what is known as the plain conical pendulum. Using the notation of Fig. 534, and equating

he opposite moments of centrifugal force and of weight, we have

$$WCN^2rh = Wq = Wr;$$

whence

$$N \propto \sqrt{\frac{q}{rh}} \propto \sqrt{\frac{1}{h}}. \quad \dots \dots \dots (347)$$

The first value in (347) is general, the second is for this particular case. In effect, we divide the rectangle  $rh$  by the arm  $r$  and get the height  $h$  of the cone of revolution, upon which the speed depends. In the lower diagram,  $h$  is laid off upon a base representing linear travel of the center  $G$ , and from it is derived a second curve which shows how the speed varies, that at the middle or 45-degree position being taken as unity.

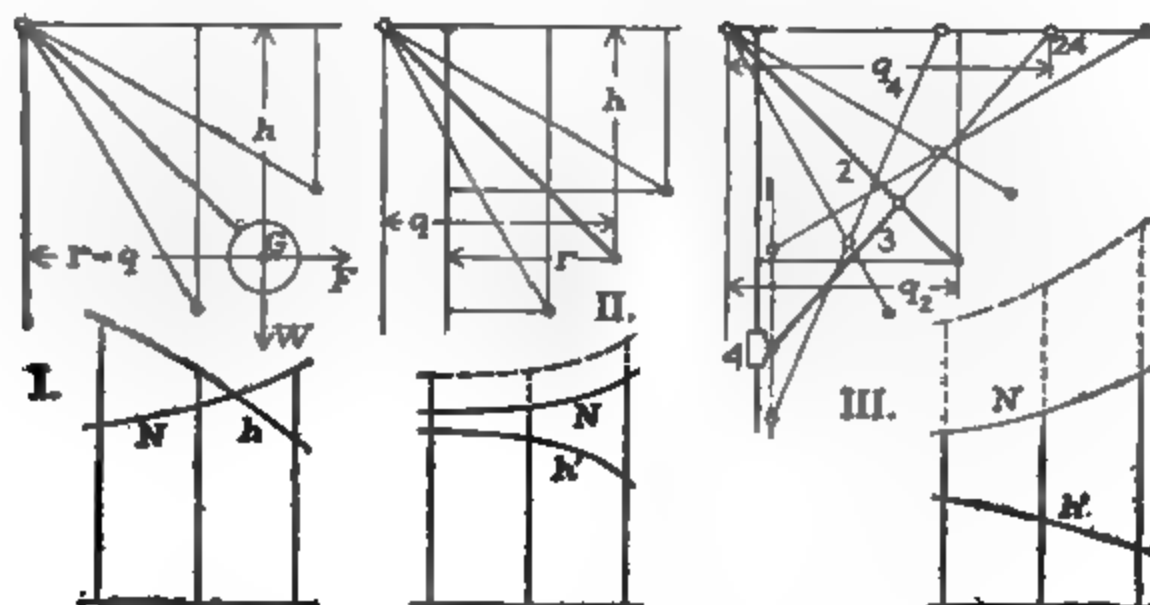


FIG. 544.—Regulation by the Fly-ball Governor.

One way of making the variation less is to offset the pivot beyond the axis of rotation, as at II. Dividing each  $rh$  by  $q$ , we get the ideal or effective height  $h'$ , which is plotted below and gives the speed-variation there shown. The full-line curve for  $N$  compares speeds just as in I.; the dotted curve represents them to the same scale as in I., showing how the decrease in the effective length of  $r$  reduces the speed at which the governor must turn for equilibrium. The lower part of the range of movement, quite a close one, is secured with this arrangement.

The effect of weight on the slide, as in Fig. 537, is shown at III. Here we reduce the weight on 4 to the arm 2, by transferring it to the instantaneous center 24. For the sake of simplicity we take  $W_4$  the same as  $W_2$ , whereupon the equation of moments is

$$W_2 CN^2 rh = W_2 q_2 + W_4 q_4 = W_2 (q_2 + q_4). \quad (348)$$

The moment-arm  $q_4$  varies even more rapidly than  $q_2$ , so that, in spite of the effect of the offset of the top pivot, the speed varies as much as in Case I. Note how greatly the actual speed (dotted curve) is increased by the addition of  $W_4$ .

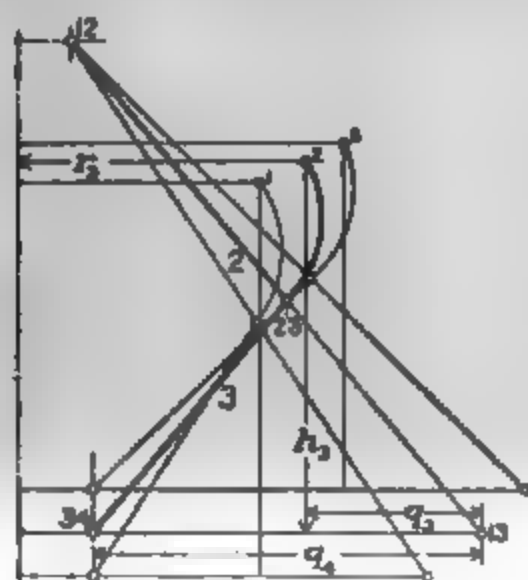


FIG. 545.—Analysis of the Proell Governor, Fig. 525 III.

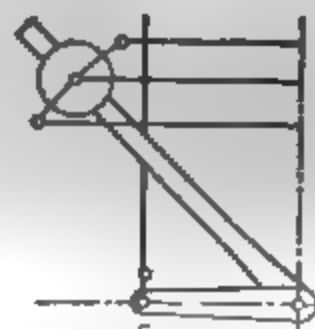


FIG. 546.—Counterpoise on Inclined Arm.

A design in which close regulation is secured by simple means through a thorough understanding of the mechanics of the governor, is outlined in Fig. 545; and the important full-size measurements are given in Table 63 J. The only dimension not taken directly from the figure is  $q_{4R}$ : to get this, the measured  $q_4$  is

TABLE 63 J. RESULTS FROM FIG. 545.

Position	$r_2$	$h_2$	$r_2 h_2$	$q_2$	$q_{4R}$	$\Sigma q$	$N$
1	6.48	10.72	69.5	4.47	11.52	15.99	4.34
2	7.73	10.11	78.1	4.70	13.42	18.12	4.31
3	8.74	9.41	82.2	4.88	14.94	19.82	4.14

multiplied by 1.285, which is the ratio of  $W_4$  to  $W_3$ ; then this reduced value can be added to  $q_3$ , for use as in Eq. (348),  $r_3 h_3$  being divided by  $\Sigma q$  to get  $h'$ . Note how, by the design of the mechanism,  $q_3$  is kept nearly constant and  $r_3 h_3$  is made to increase so as to compensate for the increase in  $q_4$ . Of course, the closer analysis which takes account of the mass of the links of the mechanism, as well as of the ball, would have to be used for getting accurate results.

An obvious scheme for varying at will the regulation of a governor like Fig. 461 is to put the poise on an inclined arm, as indicated in Fig. 546, and this arrangement is used on some governors.

With a spring for the balancing force, and the chance to vary scale, manner of elongation, and length of lever-arm, as suggested

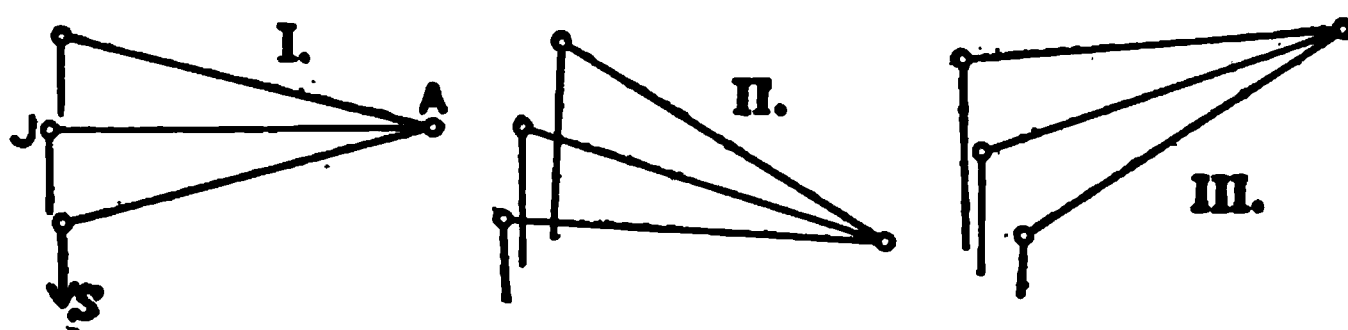


FIG. 547.—Types of Spring-connection.

by Fig. 547, we have an element of great flexibility, and the possibility of securing any regulation that may be desired.

Shaft-governors are usually provided with means of varying the speed and, incidentally, the regulation. The adjustments possible are, the addition or movement of weights, change in the spring tension, and movement of the point of spring-attachment. Changing the location of the center of gravity of the weight-arm may cause quite a change in the manner of variation of the product  $RH$ : if this is made to increase more rapidly, the regulation is made closer, and vice versa. Increasing the spring tension usually leaves its scale the same, so that the proportional increase due to deflection on account of movement of the governor is less, or again the regulation is closer. Moving the fulcrum-point of the spring in order to change the speed may or may not affect the regulation, and the effect could only be predicted after a study of the particular case.

With the fly-ball governor it is easy to provide for a constant adjustment of the speed while the engine is running. Such governors are now sometimes made with a little electric motor on the wheel, which moves the spring-attachment, and can be controlled from the switchboard—this device being used where several engines drive alternating-current generators in parallel, and one coming into service must be synchronized with the others before being thrown into the circuit.

#### § 64. The Problem of Adjustment.

(a) REDUCED MASSES IN THE FLY-BALL GOVERNOR.—In order to apply Eq. (337) to the governor in Fig. 537, it was first necessary to find  $k_2$  and  $k_3$ , which by closely approximate calculation came to 4.23" and 2.62" respectively: then  $K_2$  could be measured off once for all on Fig. 536, but  $K_3$  had to be laid off for each position. An outline of the computation for links 2 and 3 is given in Table 64 A, where all the masses are expressed in ordinary pounds. It will be noted that here again link 3 might have been left out of account without appreciable error.

TABLE 64 A. REDUCTION OF MASS TO SLIDE 4, FIG. 537.  
 $M_2 = 56.87$  lbs.  $M_3 = 0.97$  lb.  $K_2 = 18.71$ ".

Position	Arm 2			Link 3.				Total
	$a_2$	$\left(\frac{K_2}{a_2}\right)^2$	$M_{2R}$	$K_3$	$a_3$	$\left(\frac{K_3}{a_3}\right)^2$	$M_{3R}$	$M_R$
1	8.30	4.850	276.0	6.40	6.46	0.98	0.95	277.0
2	9.64	3.740	214.0	6.81	7.32	0.87	0.84	214.8
3	10.77	3.025	172.3	7.24	8.12	0.80	0.77	173.1
4	11.93	2.465	140.3	7.59	8.85	0.74	0.71	141.0
5	13.10	2.068	116.0	7.91	9.49	0.70	0.67	116.7
6	14.28	1.717	97.7	8.16	10.03	0.67	0.65	98.4
7	15.50	1.459	82.9	8.30	10.40	0.64	0.62	83.5
	1	2	3	4	5	6	7	8

For the slide 4 and the valve-gear parts that move with it, including the rods and the cam-rings but not the poise W, the



reduced mass is 54.0 lbs. The poise, in its three positions “in”, “middle”, and “out”, reduces to 40.4, 102.3, and 192.2 lbs. respectively. This gives to  $M_4$  the total values 94.4, 156.3, and 246.2 lbs. for cases A, B, and C, of which masses one-half is to be used in combination with those in column 8 in the last table. Then for case B, which we are to discuss further, the total reduced mass is given in Table 64 B.

TABLE 64 B. TOTAL REDUCED MASS, CASE B.

Position	1	2	3	4	5	6	7		
Mass	{ 355.2	293.0	251.3	219.2	194.9	176.6	161.7	pounds	1
	{ 11.05	9.12	7.86	6.82	6.06	5.49	5.03	mass units	2
1 mass unit = 32.16 lbs.									

(b) M.E.P. AND FLY-WHEEL EFFECT.—By laying out the cut-off cam-gear and locating on Fig. 465 the cut-offs corresponding to the several governor-positions, and then sketching a set of indicator diagrams, the M.E.P. curve on Fig. 548 was obtained—the operation, being analogous to that in Figs. 352 and 354, but more complicated in its initial steps. This curve is based on a boiler-pressure of 80 lbs. by gage.

Data as to the weight of the fly-wheel not being immediately available, a logical value was obtained as follows:

It is assumed that when the engine is running with an M.E.P. of the value  $P=45$  lbs., at 80 R.P.M., with the phase-ratio  $k=0.15$ , the coefficient of speed fluctuation is  $f=0.01$ . Now if the work in one revolution,  $W_R$ , is done by the force  $P$  acting through two strokes, we may take it to be approximately true that the free work  $E=0.15 W_R$  will be done by an unbalanced pressure  $\Delta P$  acting through one-half a stroke—this half-stroke corresponding to the quadrant which is the average length of a phase in the turning-force diagram. Then if  $\Delta P \times 0.5 = 0.15 \times P \times 2$ , we have  $\Delta P = 0.6P$ , here 27 lbs., as the unbalanced force which by acting through the time of one-fourth of a revolution will make a change of one per cent. in the speed of the shaft. In general, the change in momentum may be expressed as

$$\Delta P \times \Delta t = CM \Delta N, \quad . \quad . \quad . \quad . \quad . \quad . \quad (349)$$

where  $\Delta N$  is the speed-change produced by the free force  $\Delta P$  in time  $\Delta t$ , all the constant factors being included in  $C$ . Putting mass  $M$  of the wheel also into the constant, and substituting  $\Delta N = C \Delta P \Delta t$  the values

$$\Delta P = 27, \quad \Delta t = \frac{1}{4} \frac{60}{N} = \frac{15}{N} = \frac{15}{80}, \quad \Delta N = \frac{80}{100},$$

get

$$\frac{80}{100} = C \times 27 \times \frac{15}{80}; \quad C = 0.158.$$

Then for this engine we have the relation

$$\Delta N = 0.158 \Delta P \Delta t. \quad . . . . . (350)$$

(c) ADJUSTMENT BY THE FLY-BALL GOVERNOR.—The data for calculation of this action are collected in Fig. 548, several examples are fully plotted in Figs. 549 to 551, and the method is set forth in Table 64 C, which carries the numerical values of Fig. 549 through one revolution. Although here split into three sections, the table is really continuous, each calculation being continued in a line carried across all three parts; nevertheless, the division has a logical basis, because the first section is concerned with the action of the main engine-mechanism (of the shaft or wheel, more particularly), the second section shows the dynamic conditions in the governor (force and mass), and the third determines the movement of the governor.

TABLE 64 C. FORM FOR CALCULATION OF ADJUSTMENT ACTION.

Calc. No.	Time. $\Delta t$	M E P.		Speed.			
		$P$	$\Delta P$	$\Delta N_E$	$N_E$	$N_G$	$N_B$
0		30.5			81.20	60.90	60.9
1	.1843	30.5	+10.5	+.306	81.50	61.12	61.0
2	.1837	"	"	"	81.81	61.36	61.1
3	.1832	29.3	+ 9.3	+.270	82.08	61.56	61.3
4	.1827	"	"	"	82.35	61.76	61.7



TABLE 64 C.—Continued.

	<i>W</i>	<i>P</i> <sub>iso</sub>	Force. <i>F</i>	<i>l</i>	<i>l</i> <sub>m</sub>	Mass. <i>M</i>	<i>M</i> <sub>m</sub>
0	86.2	232.0	86.2	0		6.82	
1	86.2	231.7	86.8	+0.6	+0.3	6.82	6.82
2	86.1	230.8	87.0	+0.9	+0.75	6.77	6.79
3	86.0	228.8	87.0	+1.0	+0.95	6.70	6.74
4	85.7	224.9	85.9	+0.2	+0.6	6.53	6.62

TABLE 64 C.—Continued.

	Velocity.			Displacement.			
	<i>Δv</i>	<i>v</i>	<i>v</i> <sub>m</sub>	<i>ΔS</i>	<i>S</i>	Est.	<i>S</i> <sub>N</sub>
0		0			0		0
1	+ .0081	+ .0081	+ .0041	+ .0088	.009	.01	.085
2	+ .0203	+ .0284	+ .0182	+ .0402	.049	.05	.170
3	+ .0258	+ .0542	+ .0413	+ .0906	.140	.13	.245
4	+ .0166	+ .0708	+ .0625	+ .1372	.277	.28	.313

*Behavior of the Engine.*—The methods to be used in the last determination just mentioned are very much like those in the detailed analysis of fly-wheel action, § 36 (d) to (g); and it will be well to review that discussion at this point. For the engine-shaft, however, we shall not follow the periodic fluctuation in speed, but as a simplifying approximation will assume that only the change in mean speed due to an alteration of load or power need be considered. The load is supposed to change abruptly from a value represented by the mean effective pressure  $P_1$  to one represented by  $P_2$ , this event taking place just after cut-off has fixed the power which will be developed during the next half-revolution. Each calculation is made for a quarter-revolution, the same period that was used in getting (349) or (350); and the length of this period,  $\Delta t = 15 \div N$ , varies with the R.P.M. of the engine. As an additional simplification it is assumed that the position of the governor just at the end of each half-revolution will determine the power for the next half, according to the M.E.P. curve on Fig. 548; and

ADJUSTMENT DIAGRAM FOR FLY-BALL GOVERNOR IN FIG. 537.

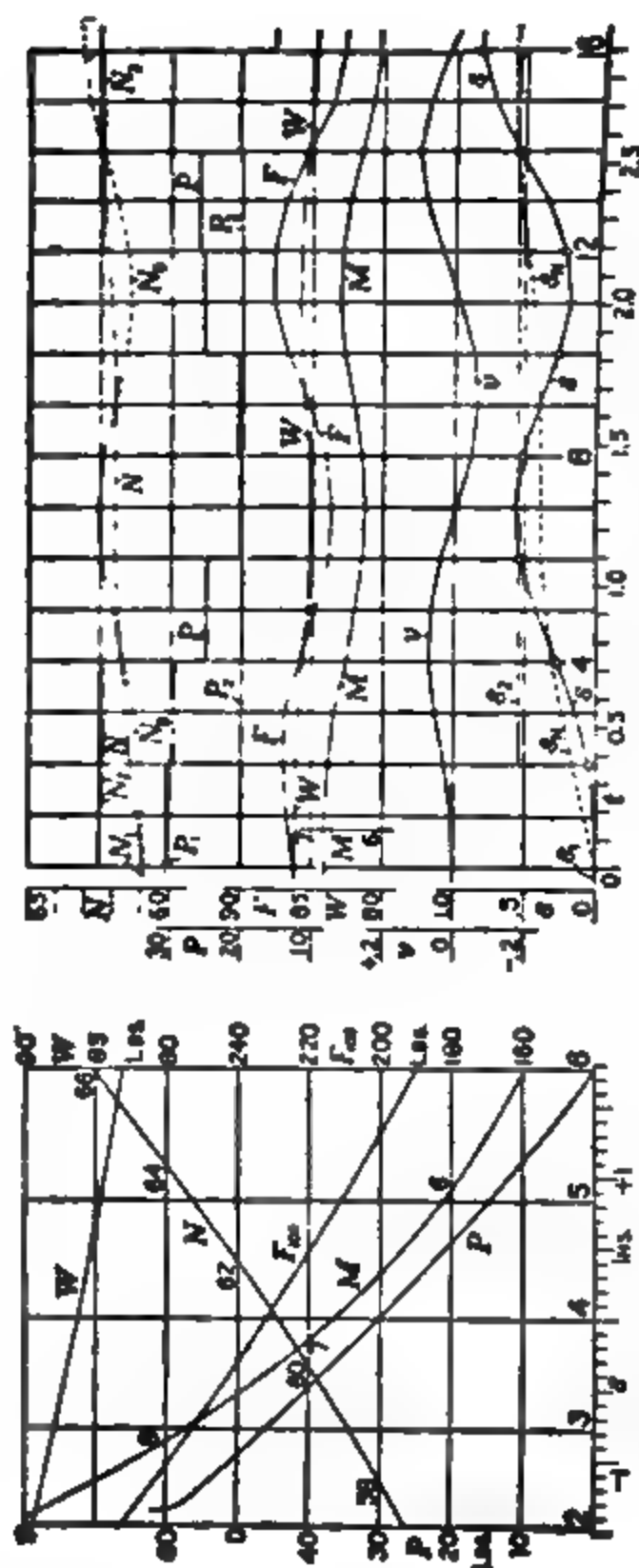


FIG. 548.—Data Curves.

FIG. 549.—Adjustment,  $P_1 = 30.6$  to  $P_2 = 20$ .

Base of Fig. 548 is path of reduction-point 34 on slide 4; forces and masses are all reduced to this point, as here-  
before designated by the subscript R. Numerical data for the several curves will be found as follows:

W — weight force,

$F_m$  — centrifugal force at 100 R.P.M.,

N — balancing speed of governor,

M — mass of governor and attached parts,

P — mean effective pressure on piston— from approximate indicator diagrams not here shown

Table 63 C, column 6.

Table 63 B, column 7.

Table 63 D, column 4.

Table 61 B, line 2.

ADJUSTMENT DIAGRAMS FOR FIG. 537, Continued.

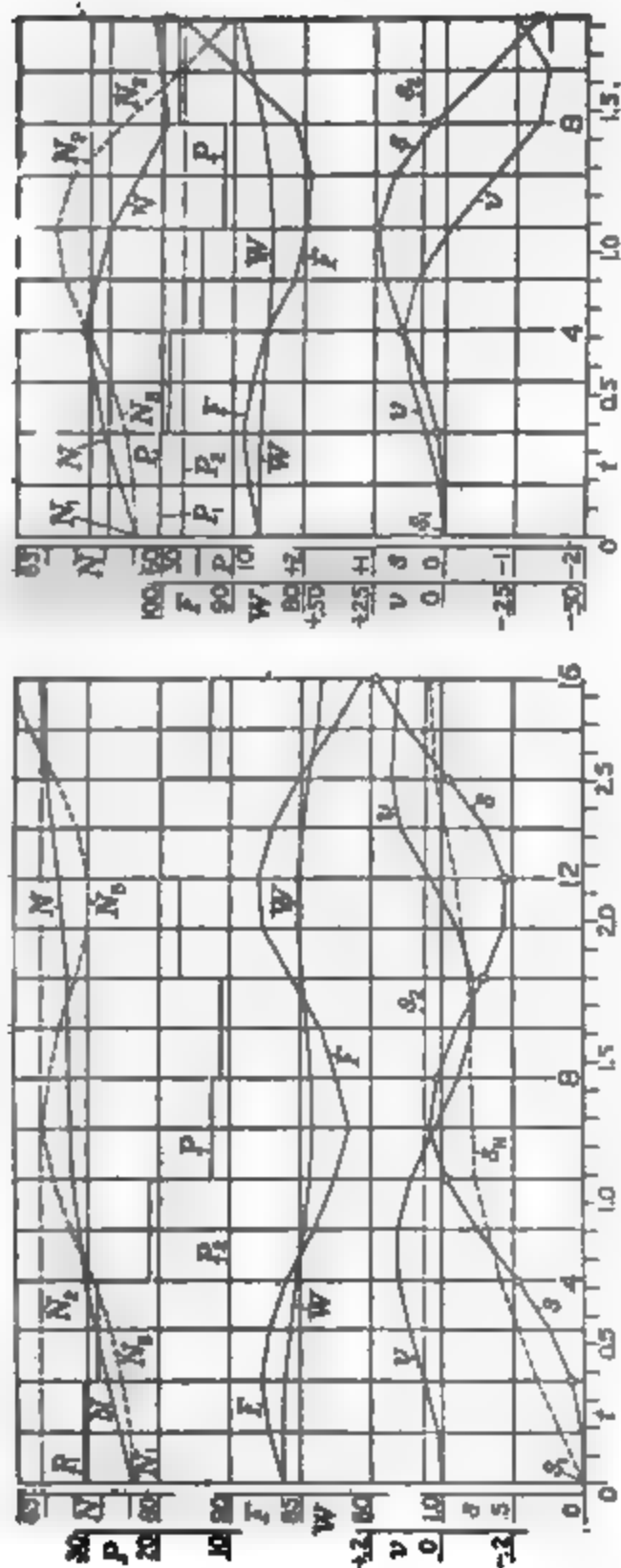


Fig. 551.—Fly-wheel Light.

Fig. 550.—Adjustment,  $P_1 = 30.5$  to  $P_2 = 10$ .

QUANTITIES AND SYMBOLS FOR ADJUSTMENT CURVES.

Mean Effective Pressure.

- $P_1$  — M.E.P. for initial load.
- $P_2$  — M.E.P. for changed load.
- $P_3$  — actual, variant M.E.P.

Rotary Speed of Governor.

- $N_1$  — initial steady speed.
- $N_2$  — final steady speed.
- $N_3$  — actual speed.
- $N_4$  — balancing speed for the governor-position at any instant.

Forces and Mass.

- $W$  — weight force
- $F$  — centrifugal force
- $M$  — mass of governor

Motion of Reduction-point.

- $v$  — velocity, feet per second.
- $s_1$  — initial position, zero at position 4.
- $s_2$  — final position, for new load.
- $s$  — actual travel, in inches.
- $s_0$  — position for balancing, at the speed existing at any instant.

All reduced to point 34.

these critical points are marked with small circles on the curve of  $s$ .

Under M.E.P. we give in Table 64 C, first the actual mean effective pressure  $P$ , then the unbalanced pressure  $\Delta P = P - P_0$ ,  $\approx P - 20$ . Substituting in Eq. (350) we get the speed change  $\Delta N$ . Here it is necessary to give both engine-speed  $N_E$  and governor-speed  $N_G$ , the latter being used when finding the centrifugal force-effect. The balancing-speed  $N_B$ , corresponding to the actual position of the governor, is found graphically from Fig. 548 after the whole calculation has been finished and  $s$  determined: its usefulness will be brought out presently.

*Forces in the Governor.*—The problem of force-action and resulting motion now before us is one that can be solved only by a trial method. We know the conditions at the beginning of any computation-period, as given, for instance, in line zero of Table 64 C for the first calculation to be made; but the values of  $W$ ,  $F$ , and  $M$  at the end of this period depend upon the final value of  $s$ , or upon the distance moved by the governor during the period. We must therefore estimate the probable value of  $s$ , measure the forces and mass from Fig. 548, run through the calculation and find  $s$ ; then if this computed value does not agree pretty closely with the estimated governor-position, new measurements must be made and the operation repeated. Only final calculations are shown in the table, the value of  $s$  under "Est." being that for which  $W$ ,  $F_{100}$ , and  $M$  are measured from the curves.

The computation of  $F = F_{100} \times N_G^2 \div 100^2$  is the only one in the whole operation for which the slide-rule is not always sufficiently accurate—although in this case slide-rule values are used. Having found  $F$ , we get the free force  $f = F - W$ ; then the mean of the values of  $f$  at the beginning and at the end of the period is the average force which acts during the period  $\Delta t$ , upon the average mass  $M_m$ .

*Movement of the Governor.*—Having force and mass, the change in velocity of the reduction-point in the time  $\Delta t$  is found by the equation of momentum,

$$M_m \Delta v = f_m \Delta t, \quad \text{or} \quad \Delta v = \frac{f_m \Delta t}{M_m} \quad \dots \quad (351)$$



For  $I_A$  the method is

$$T_A \theta_1 = I_{AR} Q \theta_1$$

$$I_{AR} = \frac{T_A}{Q} \frac{\theta_1}{\theta_1} = 0.439 \frac{\theta_1}{\theta_1} \quad \dots \quad (358)$$

For piece 3 we get  $I_T$  and multiply it by  $V_{31}$  from Table 63 E, the formula reducing to

$$I_{TR} = \frac{W}{g} \times 0.1047 \frac{R}{12} V_{31} = .00136 R V_{31}; \quad \dots \quad (359)$$

while for angular inertia we need only change the constant in (358) to .00477, as determined by the different values of  $W$  and  $k$ .

Similar substitutions give constants as follows:

for 4, .00987 in (357), while  $I_{AR}$  is constant.

for 56, .00122 in (359).

TABLE 64 D. REDUCED INERTIA FORCES IN FIG. 539.

Position.		1	2	3	4	5
Piece 2	$R_2$	16 23	16 97	17 66	18 30	18 90
$W = 32.9$	$h_2$	-3 41	-4 00	-4 51	-5.00	-5 43
$M = 1.023$	$\theta_2/\theta_1$	1 444	1 452	1 477	1 520	1 607
$k = 5.75$	$I_{TR}$	- 1057	- 1306	- 1556	- 1850	- 2180
	$I_{AR}$	- 0634	- 0637	- 0648	- 0667	- 0705
Piece 3	$R_3$	11 48	10 59	9 65	8 69	7 65
$W = 5.0$	$V_{31}$	-1 343	-1 294	-1 254	-1.216	-1 196
$M = 0.155$	$\theta_3/\theta_1$	0 000	-0 050	-0 103	-0 167	-0 261
$k = 4.86$	$I_{TR}$	- 0209	- 0186	- 0164	- 0143	- 0124
	$I_{AR}$	+ 0000	+ 0002	+ 0005	+ 0008	+ 0013
Piece 4	$R_4$	2 52	2 02	1 52	1 02	0.54
$W = 24.5$	$h_4$	-0 67	-0 28	+0 19	+0 92	+2 43
$M = 0.763$	$I_{TR}$	0017	0006	+ 0003	+ 0009	+ 0013
$k = 4.08$	$I_{AR}$	- 0165	- 0165	- 0165	- 0165	- 0165
Piece 56	$R_{56}$	11 27	10 73	10 23	9.75	9 30
$W = 4.5$	$V_{31}$	-0 897	-0 875	-0 871	-0 873	-0.898
$M = 0.140$	$I_{TR}$	- 0123	- 0115	- 0109	- 0104	- 0102
Total	$I_T$	- 1406	- 1613	- 1826	- 2088	- 2353
Reduced	$I_A$	- 0799	- 0800	- 0808	- 0824	- 0857
Inertia	$I$	- .2205	- .2413	- .2634	- .2912	- .3250

With these various coefficients the results given in Table 64 D are obtained, all of the data except  $h$  being repeated from preceding tables. As to the algebraic signs, which show whether inertia works with or against centrifugal force, it is easier to determine these by simple inspection of the mechanism than to formulate and apply general rules. It will be noted that the distinction in sign begins with the determining lever-arm  $h$  or velocity-ratio  $V_1$ . In this illustrative example all the possible forces are determined, without regard to their relative importance; a simple preliminary exercise of judgment would indicate that such small forces as  $I_{3A}$  and  $I_{4T}$  might be neglected. The final value of  $I$  is plotted on Fig. 552.

TABLE 64 E. REDUCED MASS IN FIG. 539.

Position.		1	2	3	4	5
Piece	2	3.703	3.740	3.872	4.103	4.585
Mass-	3	0.576	0.579	0.591	0.603	0.656
units	4	0.751	0.751	0.751	0.751	0.751
	56	0.239	0.216	0.199	0.183	0.181
TOTAL	$M_R$	5.269	5.286	5.413	5.640	6.173

(e) REDUCED MASS IN FIG. 539.—In this determination we follow Eq. (338), except that ratios of linear velocities must be used with pieces 3 and 56. Again, linear symbols are given on Fig. 526, certain important dimensions on Fig. 539. Stated as concisely as possible, the methods used in getting Table 64 E are as follows:

PIECE 2.  $M = 1.023$ ;  $k = 5.75$ ;  $r = 6.58$ ;  $Q = 6.75$ :

$K^2 = k^2 + r^2 = 76.46$ ;  $Q^2 = 45.56$ .

$$M_R = M \frac{K^2}{Q^2} \left( \frac{\theta_2}{\theta_4} \right)^2 = 1.777 \left( \frac{\theta_2}{\theta_4} \right)^2 \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (360)$$

PIECE 3. Since the angular velocity is so small, consider the mass as if concentrated at the center of gravity; then

$$M = 0.155, \quad M_R = M V_3^2, \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (361)$$

$V_3$  being taken from Table 63 E, line 3.

PIECE 4.  $M = 0.763$ ;  $k = 4.08$ ;  $r = 5.31$ ;  $K^2 = 44.90$ :

$$M_R = K^2/Q^2 \times M = 0.985 \times 0.763 = 0.751.$$

PIECE 56.  $M = 0.140$ ;  $M_R = MV_s^2$ .

(f) DATA CURVES FOR THE AMES GOVERNOR.—Examination will show that the curves of  $B$ ,  $F_{100}$ , and  $N$  on Fig. 552 do not quite correspond with the numerical values in Tables 63 H and I. The latter are correct for the primary data, a minor error being included in the determination of  $F$  as plotted on the figure: this is compensated by a slight change in the scale of the spring, and the only difference of any real importance is found in the fact that here the regulation is a little less close than in Table 63 I. This difference could be made by a small change in the amount of shot in the hollow head of arm 2, so that while not in absolute agreement with Fig. 539, the curves in Figs. 552 to 554 are truly representative of the action of the governor.

The rapid drop in the curve of  $P$  (the M.E.P.), which falls to zero just beyond position 4, is due to the great offset from the crank-line of the pivot-point  $P$  in Fig. 539. This makes the E-locus draw in toward the center  $O$ , so that it cuts the lap-circle before the governor reaches its outer limit. The action is like that in Fig. 360 III., but is not here determined with any great accuracy.

The fly-wheels on this engine weigh 2100 lbs., at an effective mean radius of 40.1 ins. One pound of free M.E.P. is equivalent to a total force of 113.1 lbs. on the 12-inch piston, which reduces to 10.77 lbs. at the wheel-rim (where it will directly accelerate the latter), according to the work-equation or virtual-velocity relation,

$$\Delta P_R = \frac{113.1 \times 12}{3.142 \times 40.1} \Delta P = 10.77 \Delta P. \quad \dots \quad (362)$$

In the time  $\Delta t$  this force, acting on the wheel-mass  $M = 65.3$ , will produce the velocity-change  $\Delta V$ ; which, at the end of the radius  $40.1 \div 12 = 3.34$  ft., is equivalent to a change in angular velocity



the value  $\Delta\theta = \Delta V + 3.34 = 0.1047 \Delta N$ ; whence

$$\Delta N = \frac{10.77}{65.3 \times 3.34 \times 0.1047} \Delta P \Delta t = 0.471 \Delta P \Delta t. \quad (363)$$

(g) ADJUSTMENT BY THE AMES GOVERNOR.—The form for calculation with this governor is the same as in Table 64 C, except that columns must be introduced for the angular acceleration  $\omega = \Delta N / \Delta t$ , and for the force of inertia  $f_i = I\omega$ —an example being set forth in Table 64 G. Under our ordinary assumption as to wheel-acceleration in (c), this inertia-force is constant throughout each half-revolution. Of the two curves marked  $F$  on Figs. 553 and 554, the dotted line shows centrifugal force alone, the full, stepped line adds inertia: then the free force to move the governor is included between the latter curve and the balancing-force  $B$ . It is evident that, except at the beginning of the movement, the inertia-effect is relatively insignificant.

(h) BEHAVIOR OF THE CENTRIFUGAL GOVERNOR.—We now take up a general discussion of the action represented by Figs. 549, 550, 553, and 554. The primary conditions are as follows:

The engine has been running at the constant speed  $N_1$ , under the load  $P_1$ , with the governor resting at  $s_1$  and equilibrium between the centrifugal force  $F$  and the balancing-force  $W$  or  $B$ . For the sake of simplicity, we use only the average effect of the force  $P$ , or take the turning moment on the shaft to be constant (against a uniform load-torque), with abrupt changes, when the governor is moving, at the end of each half-revolution (corresponding roughly to the cut-off). To get the most severe conditions, we make the act of adjustment begin with an instantaneous change of the load from  $P_1$  to  $P_2$ , just after a "cut-off." The important questions to be answered are, how rapidly does the governor move to the new load-position  $s_2$ , how quickly and positively does it there come to rest, and how much fluctuation in the mean speed of the shaft does it permit?

The governor requires some little time to get into motion, because of its own inertia, chiefly because a free force to be developed only as the speed of rotation changes.

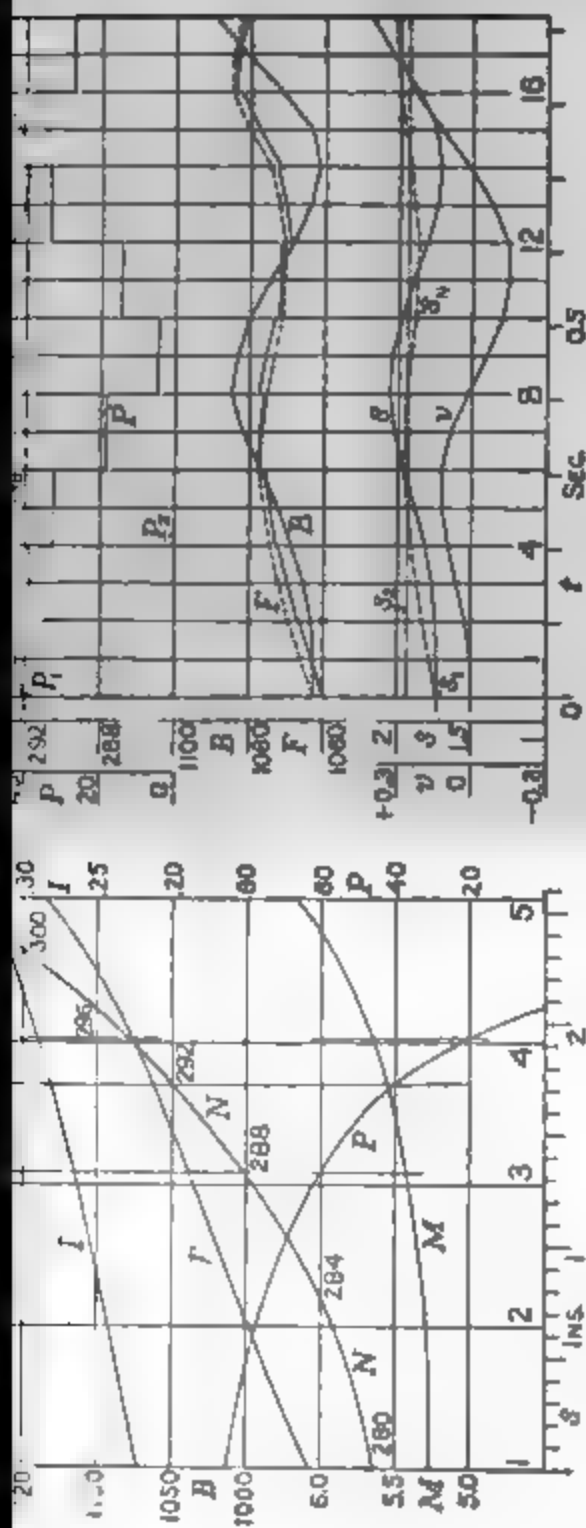
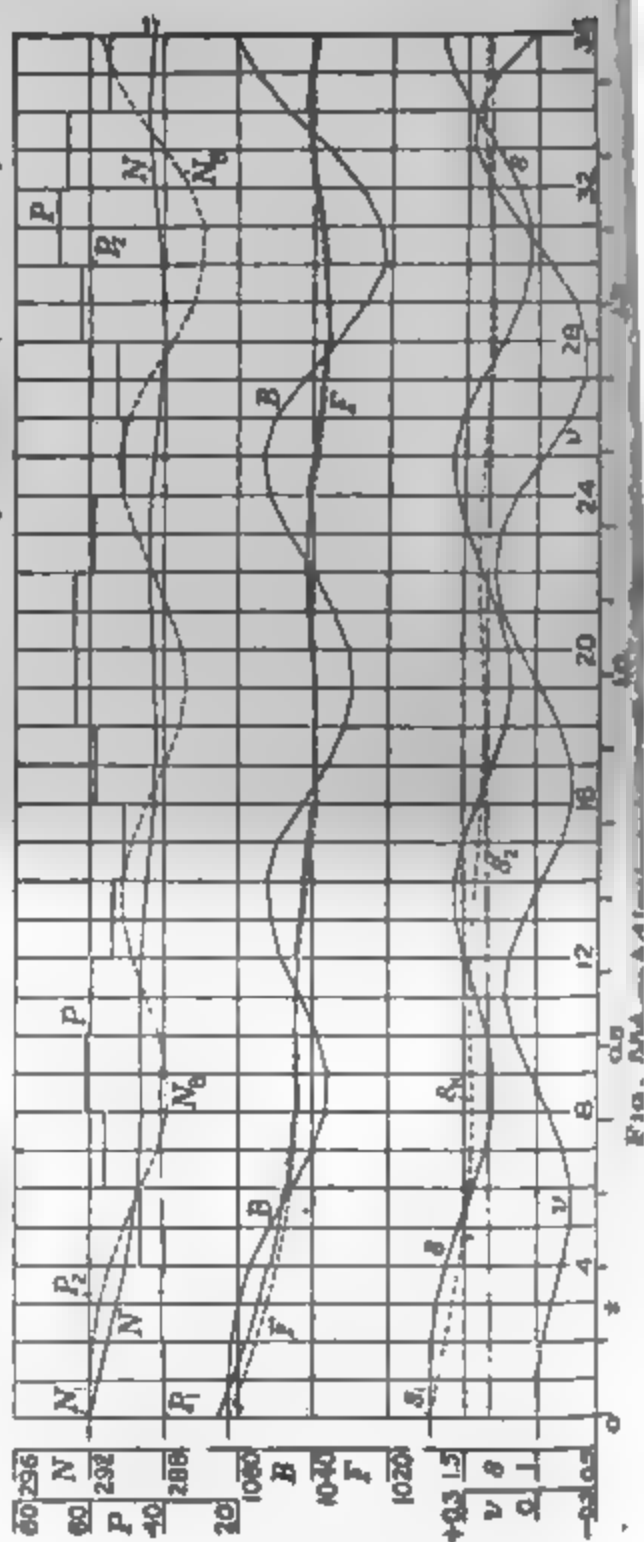


FIG. 552. Data Curves.

FIG. 553.—Adjustment A,  $P_1=40$ ,  $P_2=20$ .FIG. 553.—Adjustment A,  $P_1=40$ ,  $P_2=20$ .

ing the first revolution, then (four periods on any figure), the balanced M.E.P. is large and the engine-speed varies rapidly. The effect of this variation, as to its tendency to move the governor, is best exhibited by plotting the curves marked  $N_B$  and  $s_N$ . The curve  $s_N$  shows the speed at which, for any actual position  $s$ , the governor forces would be in equilibrium, or it is the balancing speed. Conversely,  $s_N$  shows the position which the governor would have to occupy, for any actual speed  $N$ , in order that there might be equilibrium, or it is the balancing position. The force  $f$ , active to accelerate the governor-mass, varies with the difference between  $N$  and  $N_B$ , or with the distance  $(s - s_N)$ .

As the governor acquires velocity it moves more rapidly than the engine-speed varies, presently coming to a position of equilibrium, where the curve of  $s$  crosses that of  $s_N$ , or  $N$  crosses  $N_B$ . But by virtue of the stored-up initial acceleration (or kinetic energy) it swings past this neutral point, travelling onward until checked by the reversed force  $f$ ; whereupon it swings back again, and thus  $s$  oscillates about  $s_N$  until the initial energy is absorbed by some brake-action not shown on these diagrams.

In the governor of Fig. 461, this energy-absorbent is furnished by the oil-brake or dash-pot. For the shaft-governor which we are discussing, friction in the mechanism performs the same service.

It will be noted that all the curves show essentially the same kind of action, whether we compare changes of different amplitude, as in Figs. 549 and 550, or changes in opposite directions, as in Figs. 553 and 554. But the differences in detail are greater in the latter comparison, chiefly because the inclination of the M.E.P. curve is so different on the two sides of the starting-point at  $P_1 = 40$  lbs.

(i) STABILITY IN ADJUSTMENT.—Of the four examples under consideration, the least favorable action is shown in Fig. 553. In the others, the first long swing, from  $s_1$  to the crest of the first wave beyond  $s_N$ , carries  $s$  just about to  $s_2$ , while leaving  $N$  somewhat short of  $N_2$ . Probably no simpler statement of the condition for stability in adjustment can be found than to say that the governor must move so smartly as thus to pass the engine-speed  $N_2$  and come nearly to position (to  $s_2$ ) at the end

If the first long swing, with zero friction must damp out the oscillation than any of these three examples modified by a brake-force of proportion the movement enough to let  $N$  go a little farther past  $s_2$ . But to have with full velocity, as in Fig. 553, is and shows the need of a stronger brake.

Fig. 551 supports and enforces what is stated: on account of a very light fly-wheel fourth as heavy as in Eq. (350), the fact that the governor cannot ever reach the state of zero acceleration or of rest till after  $s$  has passed  $s_1$ , while the oscillation is set up which only a check.

This brings out the important fact that the fly-wheel is a major element in determining the governor; and shows further that the smartness of the wheel in responding increases more rapidly than does so the corresponding increase in the force.

It is then the function of the governor more time to act: correlating the smartness of the governor is increased with large forces acting—as by using the center of rotation. To have a set of balancing forces insures that  $s$  shall damp down the oscillation of  $s$ , then less closer regulation can be secured with it. It must be borne in mind, however, that it is affected by irregular external forces, valve-gear.

(j) OSCILLATION OF THE GOVERNOR. The oscillation set up during adjustment, with  $b$  as in the example before us. The wheel is to  $s_2$  as a center is very nearly the same

of a swinging pendulum. For this simple case of a mass oscillating past a central position of equilibrium, under the action of a restraining force which points always toward the center and varies in intensity directly as the distance from the center, the time for a single swing is

$$T = \pi \sqrt{\frac{M}{n}}; \quad . \quad . \quad . \quad . \quad . \quad . \quad (364)$$

where  $M$  is the mass of the body and  $n$  is the ratio of the restraining force to the distance, so that the force is  $F = -ns$ . To apply this to the governor in Fig. 537, we measure on Fig. 538 the rate of divergence between curves I. and II., at position 4, and find it to be 7.5 lbs. per inch of  $s$ , or 90 lbs. per foot. Substituting, then, in (364),  $n=90$  with  $M=6.82$  (from Table 64 B), we get

$$T = \pi \sqrt{\frac{6.82}{90}} = 0.865 \text{ seconds.}$$

Measurement of the wave lengths on Figs. 549 and 550, or of the distances between points where  $s$  crosses  $s_N$ , give periods ranging from 0.84 to 0.94 seconds, showing how nearly the motion of the governor agrees with this simple, ideal action.

Under any single set of the conditions for which (364) is derived, the time of the oscillation is independent of its amplitude. But in the governor, it appears that a change which makes  $T$  longer will increase the magnitude of the extra movements, by giving the disturbing effects a longer time in which to accumulate. Since stability is secured by keeping  $s$  close to  $s_n$ , Eq. (364) emphasizes the need of a small mass in the governor, and shows further that the correlative requirement is a sufficient value of the force-ratio  $n$ . Since this sufficient restraining force can be got either by coarse regulation with small balancing forces (see Fig. 538), or by close regulation with large forces, we see the connection between stability in adjustment and sensitiveness of the governor. As the governor is to regulate more closely, the larger must be its principal forces in comparison with its mass—always bearing in

mind that the other kind of disturbing forces) calls for the present

An important fact in regard to is made evident when we note that the governor-travel  $s$  upon the  $L$  has a small oscillation similar and a little later in phase; then these relative increase in the amplitude of 554 is carried out so far especially this action; and in Figs. 549 and  $s$  swings outward in either direction from that of  $s$ , thus increasing the force  $f$  is dependent. In order that new position, the brake-force must overcome this tendency of the oscillation noted that the brake-force, in absorption, will retard the swing or increase.

As to the effect of inertia in the 553 and 554 that the small negative to check the oscillation—from which would have the opposite (undesirable)

Another oscillation of shorter but which persists continually in the effective component, along the gear reaction. This force is rough and it behaves very much like the tending to give the governor the  $s$  its mean position  $s$  that the fly-wheel uniform rotation. Detailed investigation by Professor Klein, has failed to establish stability in steady running or in adjustment by this secondary oscillation due is possible to imagine a governor be seriously affected by this action speed of rotation and a large mass influences which keep the valve-gear harmless.

The condition most destructive of stability is to have a regular variation in the load on the engine which just happens to agree interval with the period of oscillation of the governor. About the only thing that can do any good when these two get "into step" is a powerful oil dash-pot, but at the best it is a highly unsatisfactory state of affairs.

(k) BRAKE-ACTION IN THE GOVERNOR.—The need of some energy-absorbent, to bring the governor to rest at the end of an adjustment, has been made abundantly clear. The ideal brake should not at all retard the initial acceleration of the governor, but would come into action as  $s$  approaches  $s_2$ . The nearest approximation to this ideal is furnished by a good oil-brake or dash-pot, in which the resistance is due, not to thickness or viscosity of the oil, but to a proper graduation of the space for the passage of oil around or through the piston. This apparatus gives a resistance that varies as some higher power of its piston-velocity, reacting very feebly against a slow movement, but strongly checking a higher velocity of the governor.

On a vertical-axis governor, the fixed dash-pot is likely not to receive the care and attention that it ought to have. In a shaft-governor, the additional difficulty of retaining oil of the proper fluidity without introducing too much stuffing-box friction makes the use of an oil-brake rather impracticable. The designer therefore seeks for proportions and conditions which will give stability without a dash-pot: and the only brake-force that offers itself is the friction in the mechanism. A great drawback is that friction opposes just as strong a resistance to the initial movement as it does to the undesirable oscillation of the governor. The case evidently calls for a nice adjustment of force-relations, for while an absolutely frictionless governor would oscillate indefinitely, one with too much friction will make all kinds of trouble.

If we wish to determine the friction in a governor, so as to include it in the mechanical discussion of (c) or (g), the easiest way is to find by means of graphic statics all the forces, including those at the joints and bearings, making the construction without considering the pin-pressures, the pin-diameters, and the pin-velocities, we can calculate the actual, tangential

ictional resistances, and reduce the other forces by the usual method (easier and more accurate than the method of making a second comparison-circles) and comparing results.

Various designers have, as might be expected, followed different lines in developing their governors. At one extreme is an attachment to joints subjected to heaviest pressure, such as the bearing for the shaft of a very large bearing for the engine, so that its friction may act as a brake on the governor as well as for the governor as a whole. In the older Ball and Wood governor, used in the engine and 422 III.: the pendulum-piece is attached to a disk fast upon the wheel, so that the eccentric proper. The idea was that the force was greatest, a strong friction would check any movement of the governor, so that the revolution where the valve-for the governor is left free; as a result, any movement will be by jerks rather than continuous. This can be seen on the governor in the engine.

(1) CONDITIONS OF STABILITY.—The conditions of stability are set out in the last four articles as follows:

The fly-wheel should be heavy, the governor will be moderate, and the movement determined by these conditions to follow the changes in engine speed.

The governor must have a small balancing force, especially as the variation of the steady speed with the engine speed.

There should be such a relation between the initial movement will be a small one, or as these movements would be under the influence of the brake-force.



Some brake-force is essential, to damp out the oscillation which is set up in any adjustment-action. While friction performs this function in most governors, it is highly important that this friction shall not be, or become, excessive.

The preceding statements refer chiefly to stability or positiveness in adjustment under a change of load. For steadiness under the periodic fluctuating forces (within the revolution), of which the valve-gear reaction is by far the most important, high speed of rotation and a sufficient mass in the governor are the chief requirements. If the valve has a high resistance, or one that is liable to sudden great changes, the difficulty of driving it by a shaft-governor increases rapidly and may become prohibitive.

It would be possible, by further research and by a mathematical treatment of the subject, to reduce some of these requirements to more or less approximate quantitative expression—several equations of stability having been derived and published by different investigators. For present purposes, however, it is enough to have brought out clearly the general principles involved. It may be remarked that in its deeper detail the question of governing is more markedly a subject for the specialist than is probably any other matter in connection with steam-engineering.

(*m*) THE ONE-PIECE INERTIA GOVERNOR.—To get some idea of the force-action in a governor of the type in Fig. 531, we shall now analyze the example outlined in Fig. 555, which is not an

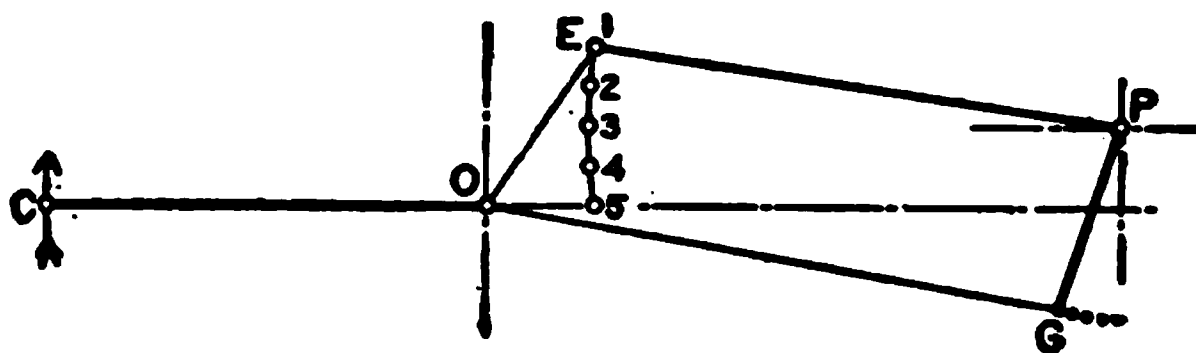


FIG. 555.—Outline of Governor of the Rites Type.

actual design, but is simply assumed for illustration. All that need be expressed graphically, of the single moving piece, is the eccentric-pendulum PE and the center of gravity at the end of a radius PG. Everything else is contained in Table 64 F.

$$I_{CQ} = I_{CQ1} + I_{CQ2} = 2.5 + 2.5 = 5.0 \text{ mA}$$

$$I_{BQ} = I_{BQ1} + I_{BQ2} = 0.5 + 0.5 = 1.0 \text{ mA}$$

$$V_{CEQ} = V_{CC} - I_{CQ} R_{C1} - I_{CQ} R_{C2} = 10 - 5.0 \times 1.5 - 5.0 \times 1.5 = 2.5 \text{ V}$$

$$V_{BEQ} = V_{BE1} = V_{BE2} = 0.7 \text{ V}$$

$$I_{EQ} = I_{CQ} + I_{BQ} = 5.0 + 1.0 = 6.0 \text{ mA}$$

Parameter	1	2	3	4	5
$I_{CQ}$	1.0	1.5	2.0	2.5	3.0
$I_{BQ}$	0.2	0.3	0.4	0.5	0.6
$I_{EQ}$	1.2	1.8	2.4	3.0	3.6
$V_{CEQ}$	8.0	7.5	7.0	6.5	6.0
$V_{BEQ}$	0.7	0.7	0.7	0.7	0.7

As the value of the collector load resistance  $R_C$  increases, the value of  $V_{CEQ}$  decreases and the value of  $I_{CQ}$  increases. The value of  $V_{CEQ}$  is maximum when  $R_C = 0$ .

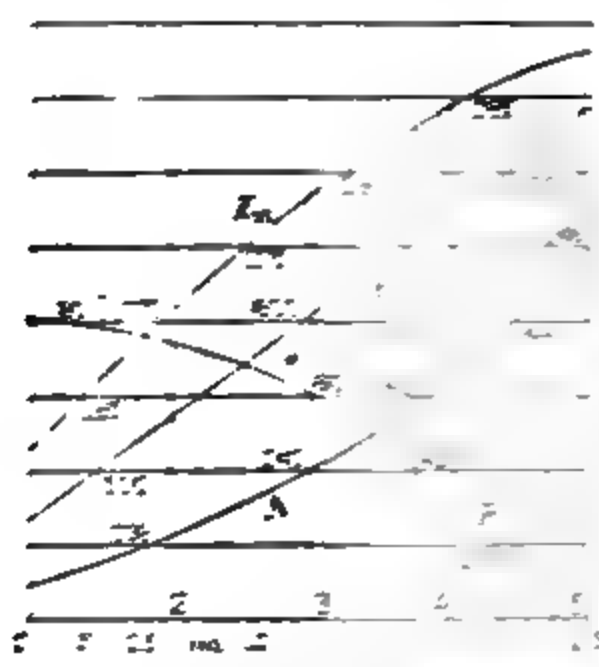


Fig. 556 - Load line

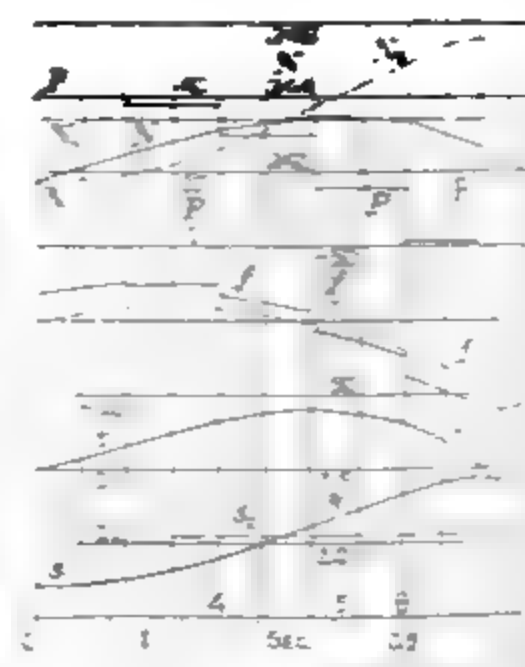


Fig. 557 - Adjusted load line

When a signal is applied to the base of the transistor, the value of  $I_{BQ}$  varies and the value of  $I_{CQ}$  varies.

When a signal is applied to the base of the transistor, the value of  $I_{BQ}$  varies and the value of  $I_{CQ}$  varies.

When a signal is applied to the base of the transistor, the value of  $I_{BQ}$  varies and the value of  $I_{CQ}$  varies.

incorporated into the design. Since mass and inertia-effect constant, only four curves need be plotted on Fig. 556. fly-wheel effect is expressed by the formula  $\Delta N=0.3 \Delta P \Delta t$ , which makes the wheel relatively heavier than with Fig. 539. The method of calculation is outlined in Table 64 G, to show it differs from that in 64 C; but the third section, which remains unchanged in form, is not repeated. Under "Speed" have the additional item  $\omega=\Delta N/\Delta t$ =here  $0.3 \Delta P$ , to be used in calculating the inertia force  $f_2=I\omega$ . The angular acceleration is expressed in R.P.M. of change per second, and since  $I$  is calculated and reduced for one unit in this measure, we get the force  $E$  in pounds by the multiplication just indicated. After the centrifugal force  $f_1=(F-B)$  has been determined and averaged over the period,  $f_2$  is added to it in order to get the total force  $f$ .

TABLE 64 G. FORM FOR CALCULATION OF ADJUSTMENT, WITH INERTIA.

Calc. No.	Time. $\Delta t$	M.E.P.		Speed.			
		$P_E$	$\Delta P$	$\Delta N$	$\omega$	$N$	$N_B$
0		40.0				241.75	241.75
1	.0618	40.0	+20.0	+.370	+6.00	242.12	241.80
2	.0618	"	"	"	"	242.49	241.92
3	.0617	37.9	+17.9	+.331	+5.37	242.82	242.20
4	.0617	"	"	"	"	243.15	242.57

TABLE 64 G.—Continued.

$B$	$F_{100}$	Forces.			$I$	$f_2$	$f$
		$F$	$f_1$	$f_{1m}$			
1654.4	283.10	1654.4					
1656.0	283.20	1660.0	+4.0	+2.0	3.63	+21.8	+23.8
1658.7	283.41	1666.5	+7.8	+5.9	"	+21.8	+27.7
1665.3	283.93	1674.0	+8.7	+8.3	3.63	+19.5	+27.8
1674.3	284.58	1682.4	+8.1	+8.4	"	+19.5	+27.9

On Fig. 557 we plot, not the total balanced force  $f$ . As in Figs. 4 and 5 shows  $f_1$ , the broken full-line curve ordinate from the zero-line.

The most important fact shown is the considerable inertia-force that is introduced when the acceleration of the engine is slow at catching up with the speed  $s_1$ . There is therefore a positive inertia force so that the governor reaches "positive" position swings away past the desired stop position an excessive oscillation. The reaction of the mechanism, is the very inertia of the free forces acting. Inherently, the wheel, or of the governor as a whole, has the radial inertia which we call on account of angular acceleration that can be produced by all sufficient weight. It is also a fact that the governor that the centrifugal effect appears from the location of the center of mass. Since therefore no reversed force is introduced until after it has passed the new limit free M.E.P., there must be a velocity introduced, with the drawback that the inertia is partly neutralized. It is difficult to say just how the governor effects the change of speed with which it is generally credited.

It is of interest to note that, as the inertia—designated above as  $I_A$  or  $T$ —of the governor-arm, the disposal of the mass of the engine. Turning from reduced mass back to the most it can do on account of its inertia, it can rotate at constant velocity while the engine is in motion. In our method of calculation, we get the inertia of the piece to acceleration with the wheel as the active force to accelerate the arm and the result that, so far as torque in

angular velocity of this body remains constant. Giving it the usual form of a long bar with heads, or making its "moment of inertia" large, simply increases its resistance to acceleration by other forces.

The example in Fig. 555 has rather poor proportions, because the center of gravity is too far from the shaft center *O* and too near the pivot *P*. To get the same effect with a small pin-pressure, *P* is brought nearer to *O*, and *G* with it; and to compensate for the decrease in *R* the lever-arm *H* is made longer. With the proportions here used the friction at the pin, with an ordinary solid bearing, would be equivalent to from 15 to 20 lbs. when reduced to the path of *E*; so that it is not hard to see where any desired amount of brake-force can be got, even without the actual brake which is sometimes used.

### § 65. Various Special Regulators.

(a) COMBINED SPEED AND PRESSURE GOVERNOR.—An air-compressor governor like that in Figs. 222 and 223 is shown in detail by Fig. 558. In its mechanism for regulating speed, it is a good representative of a large class of throttling governors, used on small engines in various lines of service: but the addition of an apparatus for controlling by air-pressure calls for some special points in the arrangement, to meet the requirement that either device shall be able to act on the valve, independently of the other.

The main mechanism, consisting of pulley-spindle 1, hollow governor-spindle 2, and fly-balls 3, 3, is self-evident. Through the rod 4 and the intermediate pieces 5 and 6, the valve 8 is pushed down when the balls fly out, throttling the steam. This valve, being of the piston rather than of the double-seated disk form, is perfectly balanced, so that the very light stem 7 is sufficient: and the cross-bar below the valve insures that it shall never fall so far as to admit steam when it is intended to be closed. Piece 5 is held against 4 by the first spring, acting through lever 9; similarly against 6 by lever 12, the springs both having a determining the running-speed. From the top

6 a small pin or spindle project  
keep these parts always in line.

The pressure-governor consists  
the air-pressure at C, with the pluz

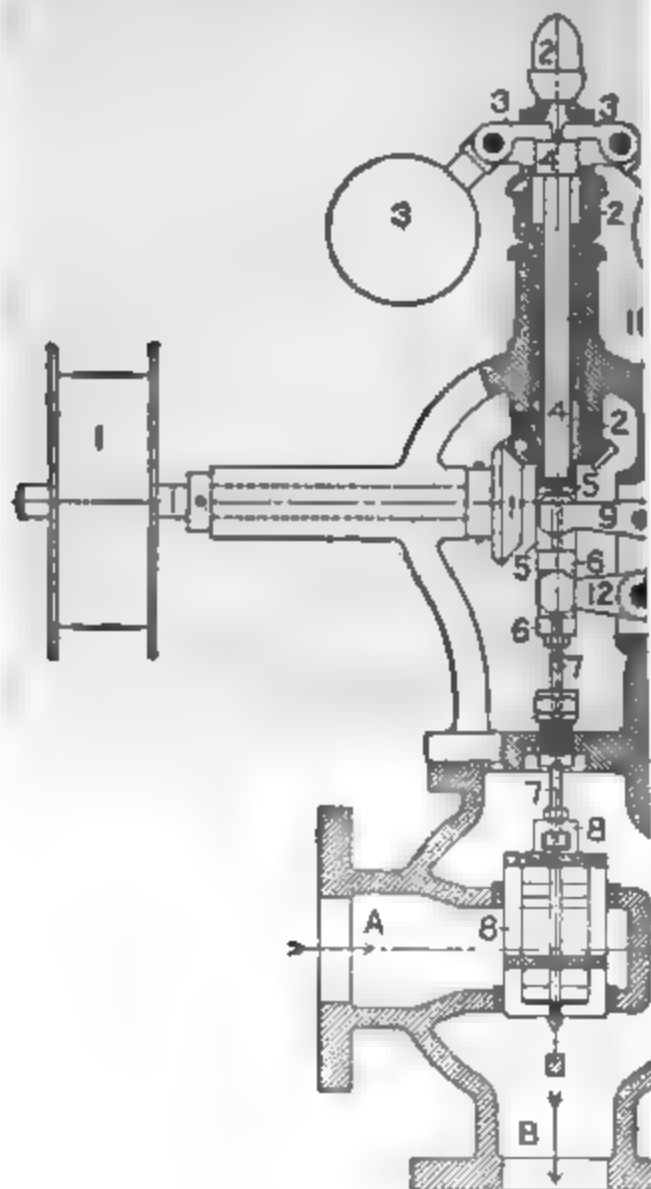


FIG. 558.—Gardner Governor

15 which is pivoted at D. When the  
maximum and raises the weight-arm  
end of 16 pushes against the outer  
valve. The stop-screw 19 can be  
quite shut off by this action, for if  
stopped, it may be in a position from  
steam is again admitted.

This last possibility is much greater with a single machine like Fig. 222 than with a duplex compressor. To eliminate such source of annoyance, a device called an "unloader" is used when circumstances call for it. Actuated by the pressure of the discharged air, this apparatus removes the load from the compressor-piston at the same time that steam is nearly shut off from the steam-cylinder. One scheme is to close the air-inlet, so that the piston moves back and forth against a low variable pressure; another opens some of the discharge-valves at both ends, so that equal high pressures are maintained on both sides of the piston. In either case, the machine continues to run at moderate speed, ready to respond at once to a drop of pressure in the air-receiver.

(b) SAFETY STOPS.—The ordinary non-automatic safety-stop for an engine with releasing valve-gear is well typified by the simple device shown on Fig. 461; but any such stop-ring or stop-pin arrangement has the decided disadvantage that it depends upon fallible human nature for being put into the service-position, and that when there it renders the engine liable to a summary shut-down by a sudden overload. Since breakage of the driving belt is the most frequent occasion of governor-stoppage, the usual scheme for making the safety-device automatic is to connect it to a pulley which rides on the governor-belt as in Fig. 559. At I. is sketched a stop S which is normally in position to support the arm L on the governor-slide: through the rod R it is connected to a lever like 2 in drawing II., so that dropping of the pulley will throw S to the left, and let the governor come all the way down.

In Fig. 559 II., the fall of pulley 3 moves the cam-rings directly, without waiting for the governor to slow down. As better shown at A, the bent lever 6, from the governor, takes hold of the rocker 5 with a spring-catch. Against the push of the trip-arms upon the cut-off cams (shown in direction by the arrows), 6 pushes solidly on 5; but if the pulley drops, the slotted bar 4 will easily pull the pin on 5 out from under the spring on 6, and rotate 5 in the left-hand direction, or that for earliest cut-off: so that here the same cams serve both for ordinary cut-off and to prevent valve-opening in an emergency. Further, since a moderate pull on one

of the governor-rods will thus release the cam-system from governor control, the engine can be quickly shut down by hand if occasion arises.

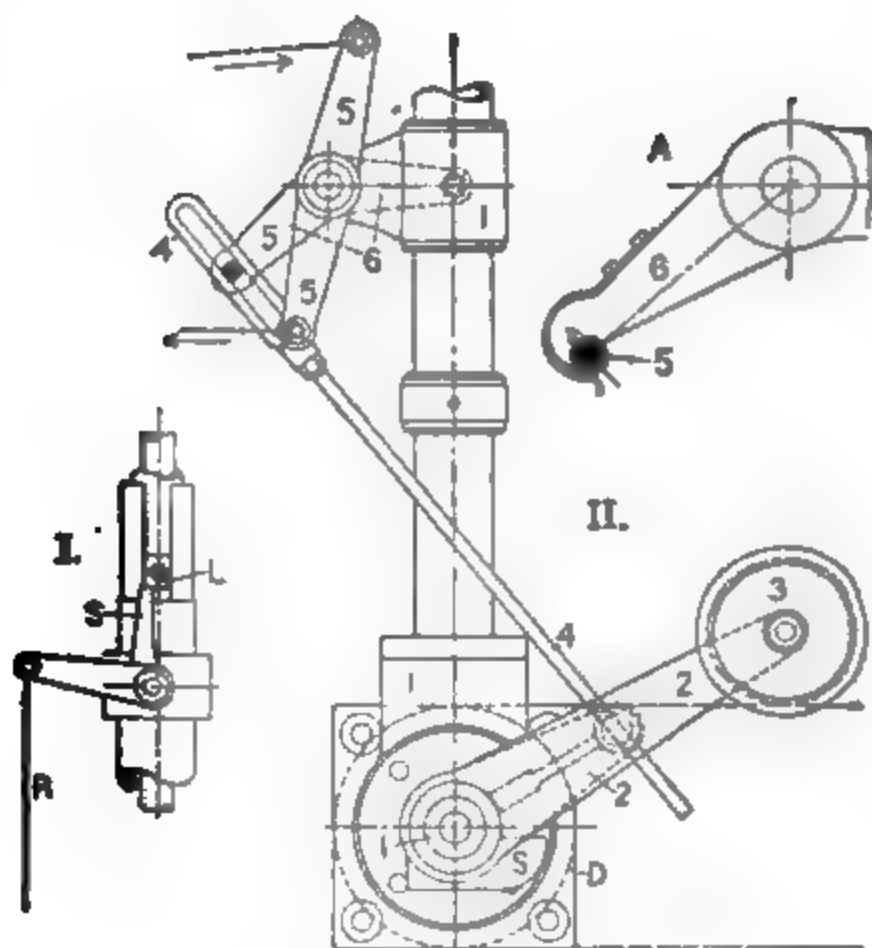


FIG. 559.—Safety-stops for Corliss Engines.  
II. Reynolds-Corliss (Allis-Chalmers).

Another type of speed-limit consists of an extra stop-valve above the main throttle, controlled by a special governor—see Fig. 219. The valve, usually of the oscillating plug type, like a Corliss engine-valve, is held open by a ratchet-detent, against the pull of a weight upon an arm keyed to the valve-spindle. If the speed rises to the maximum for which the second governor is set, the ratchet is released and steam is shut off instantaneously. The stop-valve is arranged to be thrown by hand also; and if with it there is a vacuum-breaker, to open the condenser to atmospheric pressure, the fullest control for quick stoppage is secured.

Similar to the last device in general principle is the independent safety-stop, which can be mounted on any engine, to shut the



main throttle-valve in emergency. In one quite common device, a weight is hung on a band wrapped upon a small drum; the spindle of this drum is connected by sprocket-wheels and a chain to the valve-spindle; and the drum is held by a ratchet-pawl which can be thrown out by means of an electro-magnet. Push-buttons, located wherever convenient, make it possible to stop the engines from various points in the engine-room or in the factory.

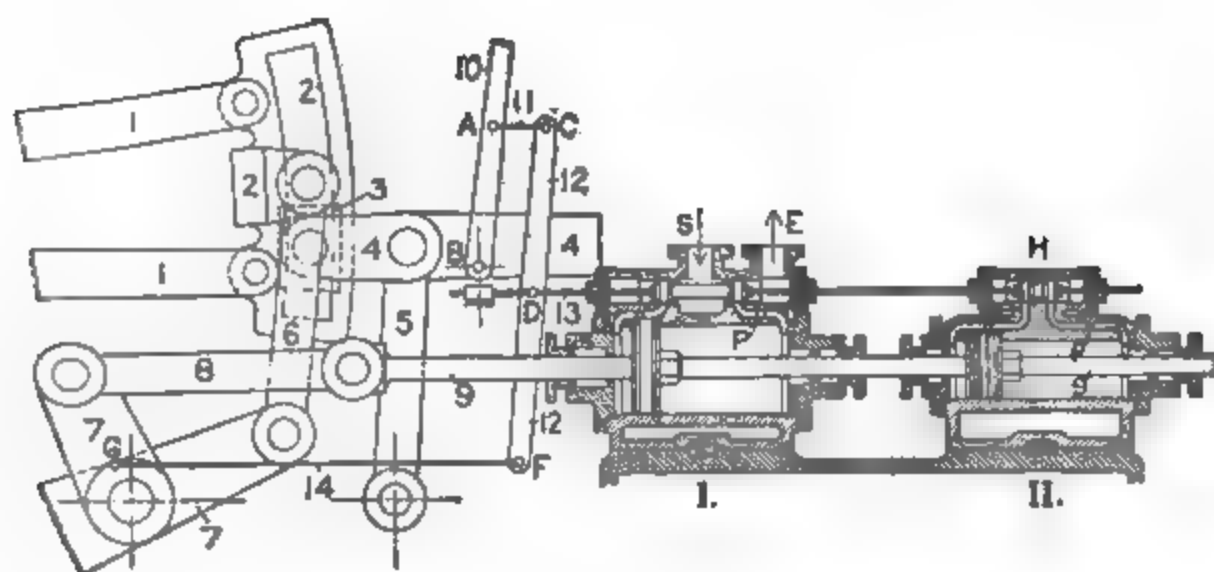


FIG. 560.—Reversing-gear for Engine in Fig. 221.

(c) REVERSING-GEARS.—On large reversing-engines, as in rolling-mill, hoisting, and marine service, the link-motion must be moved by mechanical power; the first class mentioned presents the most severe conditions, because the engine has to be quickly reversed and at very short intervals. The gear belonging to Fig. 221 is shown in Fig. 560: it is moved by the steam cylinder I. and held by the hydraulic cylinder II. The main valve-gear, pieces 1 to 6, needs no description; rocker 7 is heavily counter-balanced at the left. The secondary valve-gear begins with the hand-lever 10, pivoted at B, which moves the valve-rod 13 by means of the floating lever 12. This gear is of the self-centering type; as lever 10 is thrown to one side or the other, it carries point C with it, moving the small piston-valves; but as the piston on rod 9 moves the rocker 7, the rod GF brings back the lower end of lever 12 so as to return the valve to mid-position. The farther 10 is moved, the farther will the pistons move before the valves

me to the holding-position. Cyl  
sh-pot, to keep the mechanism  
ring it more or less quietly to res  
keep it in the desired position—for  
is not effective. In the valve-chan  
the ends and the middle space must  
the valve simply controls the flow o  
than water) from one end to the ot

A typical marine-engine, direct-  
illustrated in Fig. 561, the view inc  
and its valve-gear. Movement o  
(located as most convenient) is trans  
2, pivoted at A. This lever, acting u  
3, moves the valve-rod up or down.  
descends, it turns the rod 5 by mea  
and thereby screws 5 through 4 so as  
pins on H heavy rods run to a cri  
above H is a hydraulic cylinder whic  
desired position.

In many marine engines, the lin  
a little engine, which acts through a  
latter "self-locking" mechanism, r  
but the self-centering idea must no  
valve which admits steam to the sma

(d) INDIRECT GOVERNING.—Fig.  
arrangement used where the reaction  
that a very powerful governor would  
disturbing force. The particular dev  
engine with the Joy valve-gear, the  
m. The governor proper merely co  
chamber detailed at A, this valve det  
discharge of water to and from the  
cylinder moves, while the plungers P,  
of the whole mechanism, including  
the valve-gear, can easily be traced  
tions, it is necessary to cross the pip  
the plungers, top port going to botto

Valve (at A) is made with small positive laps, as must be the case in any such device when the function of holding predominates over that of moving.

In very large engines of the releasing-gear type, as Fig. 219 for instance, this same general method is used, the governor acting through a small hydraulic cylinder. It is necessary, of course,

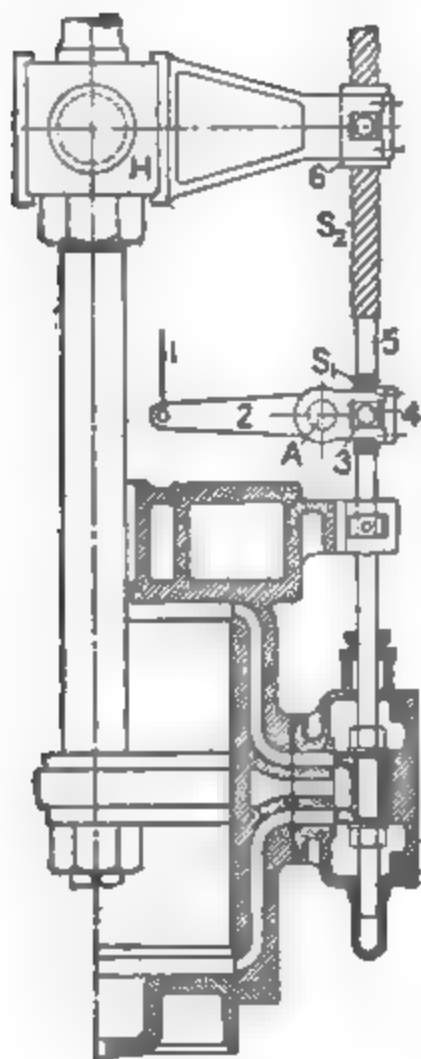


FIG. 561.—Part of Brown Reversing-gear, for Marine Engines.

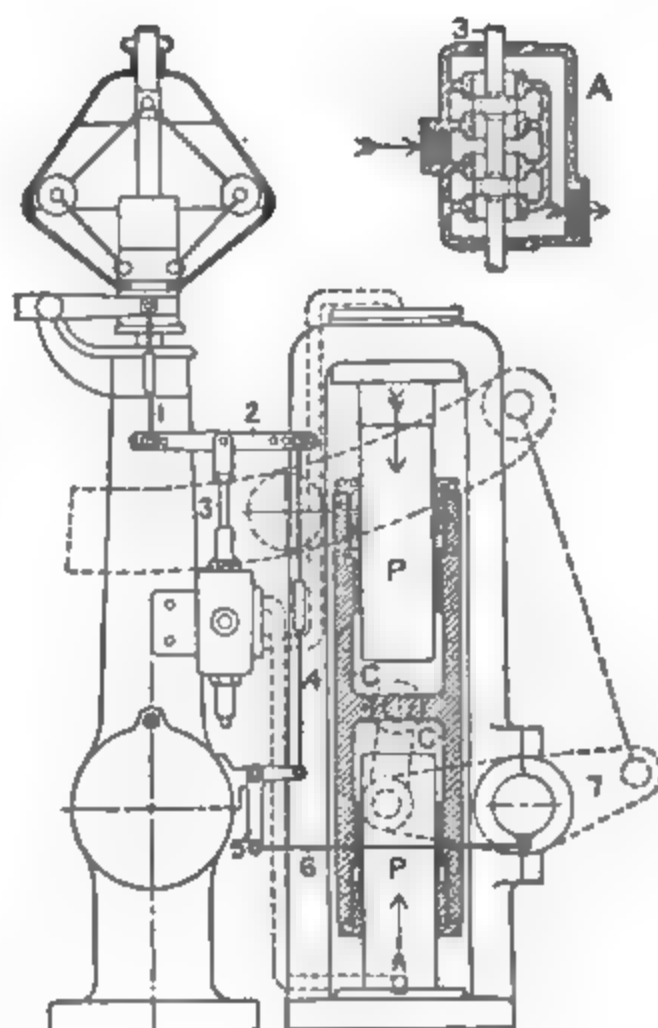


FIG. 562.—Governor for Van Vleck Engine, with Joy Valve-gear.

to have a supply of water under pressure to furnish the motive power for the actual work of adjusting the valve-gear. With a light, quick-running governor, this whole apparatus need be no more sluggish than the heavy governor that would be necessary for direct control.

tion; that is, 1 will rise until the short edge of the tooth just slips out of the slot; as soon as 7 reverses, it will engage 6 with the longer face of the tooth, and turn the screw back.

As to downward movement of 1, zero cut-off, or failure to open the valve, is reached before 1 gets to the bottom, so that there is no need of a releasing-device. In this direction acts the safety-valve for speed, intended to prevent a runaway in case of a sudden failure of load, as by the bursting of a pipe. Block 1 is not directly threaded for screw 5, but the thread is formed only on the part of nut 2, held between the caps 3 and 4. Wedge 16 is connected to the piston of a small cylinder carried on 1—see Fig. 564 I. A fly-ball governor is so arranged that when the limit of speed is reached, the weight W, Fig. 564 II., will be released from a catch

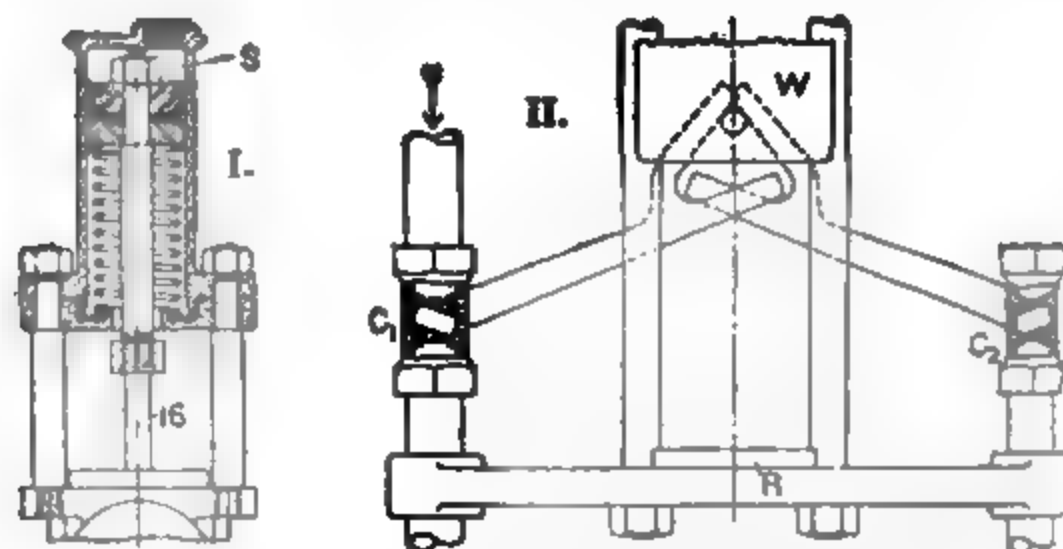


FIG. 564.—Detail of Speed Limit Governor, Snow Gear.

which holds it up and will fall to the cushion R. In falling it opens the cocks  $C_1$  and  $C_2$ . The pipe to  $C_1$  comes from a main drain, so that it contains water under the boiler-pressure; opening  $C_1$  admits this pressure to the cylinder S, pushes down the wedge 16 and lets block 1 drop to its lowest position, shutting off steam from the engine. At the same time, opening  $C_2$  admits air to the condenser, "breaking" the vacuum. Connection to cylinder S is made by means of swivel joints in the pipe.

(f) GOVERNORS FOR MARINE ENGINES.—The marine engine does not need an automatic governor to regulate its ordinary running, because it generally works against a steady

and receives close and constant oversight. What it does require is some device that will control a sudden burst of speed, as when pitching of the ship raises the propeller partly out of the water, and that will act as a safety-stop if shaft or propeller breaks. The ordinary fly-ball governor is not well adapted to this service, since it will be disturbed by motion of the ship, and must control indirectly, through a more or less sluggish mechanism: although an effective safety-stop can be secured by fastening a ratchet-toothed ring upon the engine- or propeller-shaft, to drive a mechanism which will close the throttle-valve when a centrifugal governor throws in the pawl. Attempts have been made to govern by means of the variation in external water-pressure at the stern or by the angular position (fore and aft) of the ship—the latter by fitting up a long horizontal pipe with risers at the ends, like a gigantic U tube; as the stern rises, water in this pipe will rush forward and might be made to operate a governing mechanism. It need hardly be said that these devices have failed to meet the requirement of promptness and reliability.

The Aspinall governor, now to be described, is probably the best marine governor that has been devised. The general scheme is shown in Fig. 565,

where the governor is seen at G, mounted on an oscillating lever AB driven by the cross-head—the air-pump lever being used when the pumps are directly connected. The governor acts upon the lever L, to open or close the special “butterfly” throttle-valve V. The intermediate lever H is an essential part of the gear, because it makes possible

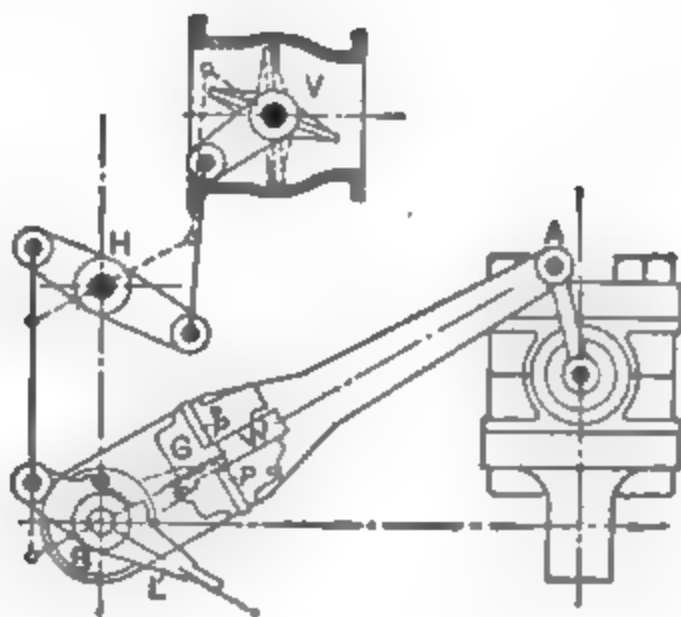


FIG. 565.—General Arrangement, Aspinall Governor for Marine Engines.

balance of the rods. Very often the governor is with a hand-control system.

The construction of the governor is fully shown in Fig. 566 where the working parts are in mid-position; its action can be

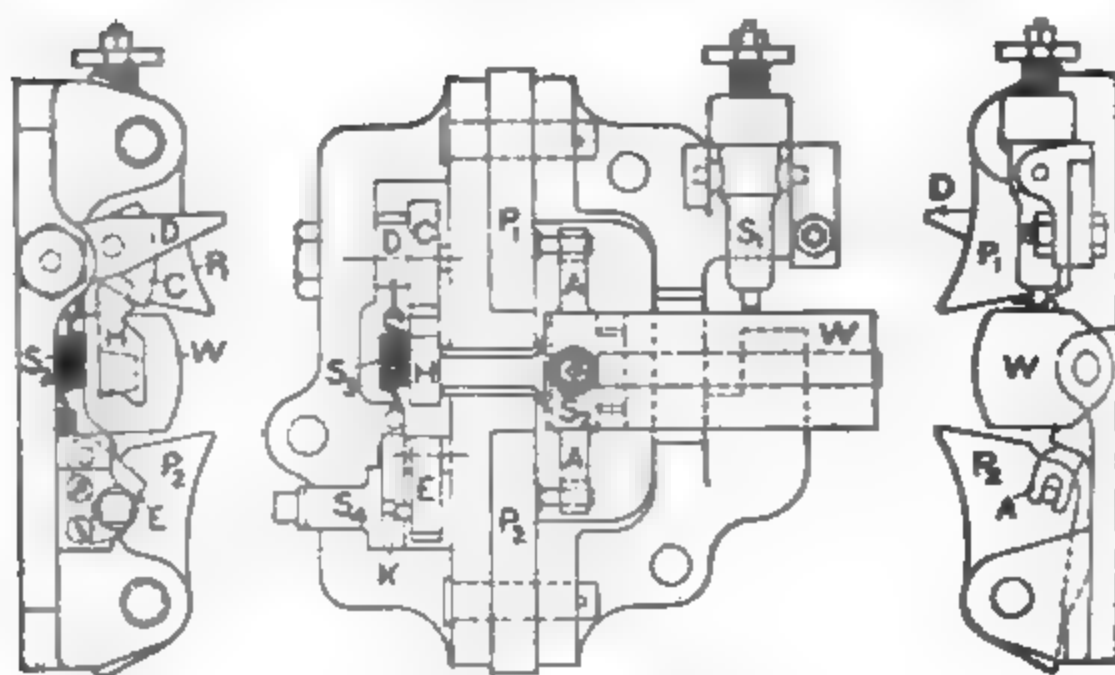


FIG. 566.—Aspinall Governor.

better understood with the help of Fig. 568. The normal-running position is given at Fig. 568 I., the weight W being down and the

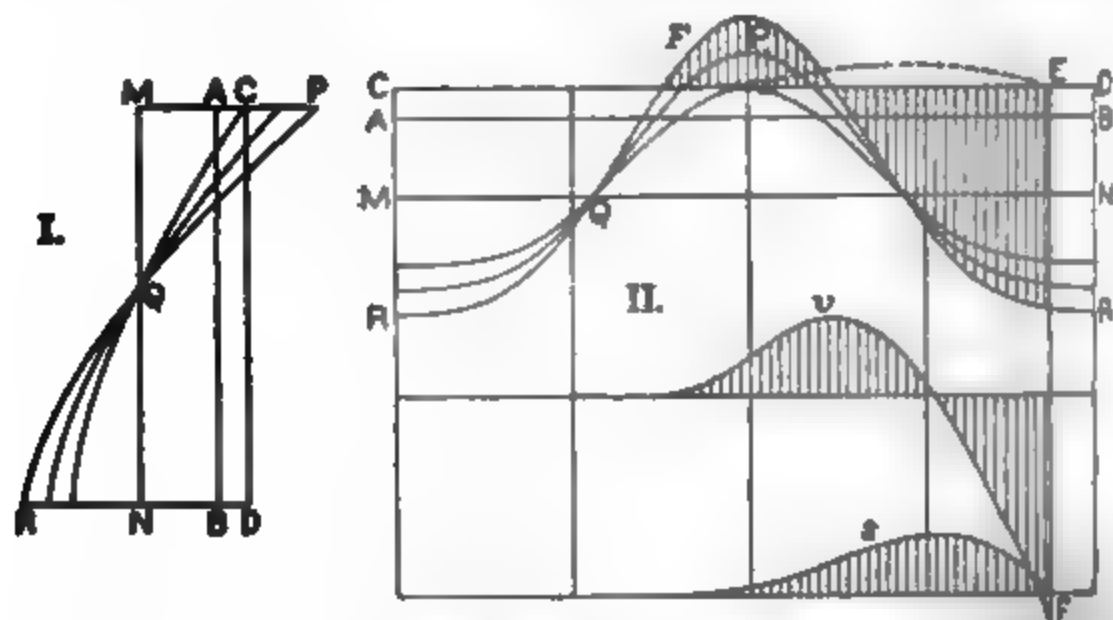


FIG. 567.—Dynamic Action of the Aspinall Governor.

upper pawl  $P_1$  out: then lever L remains down, as in Fig. 568, and the valve is open.

The governor as a whole has very nearly the same vertical reciprocating motion as the engine-piston (but reduced in magnitude); so that the weight  $W$  is subject to an acceleration and an inertia-action just like that of the piston. This is diagrammed in Fig. 567, on the stroke-line at I., on the developed-circle or time base at II.—refer to Figs. 125, 129, 169. The weight  $W$  is held down by its own gravity-force, laid off as  $AM$ , and by the spring-force  $CA$ , exerted by the spring  $S_1$ . Then only when the inertia-curve rises above the line  $CD$  is there an unbalanced upward force to lift  $W$  from the bottom stop.

The curves of velocity  $v$  and displacement  $s$  (of  $W$  with respect to the body of the governor) are derived from the force-curve by the methods used in Fig. 141, which are essentially the same as in the determinations of governor-adjustment in this chapter. Intended only for illustration, these curves are drawn without taking account of the increase of spring-force at  $S_1$ , due to the lifting of  $W$ , and shown by the dotted curve above  $CD$  in Fig. 567 II.

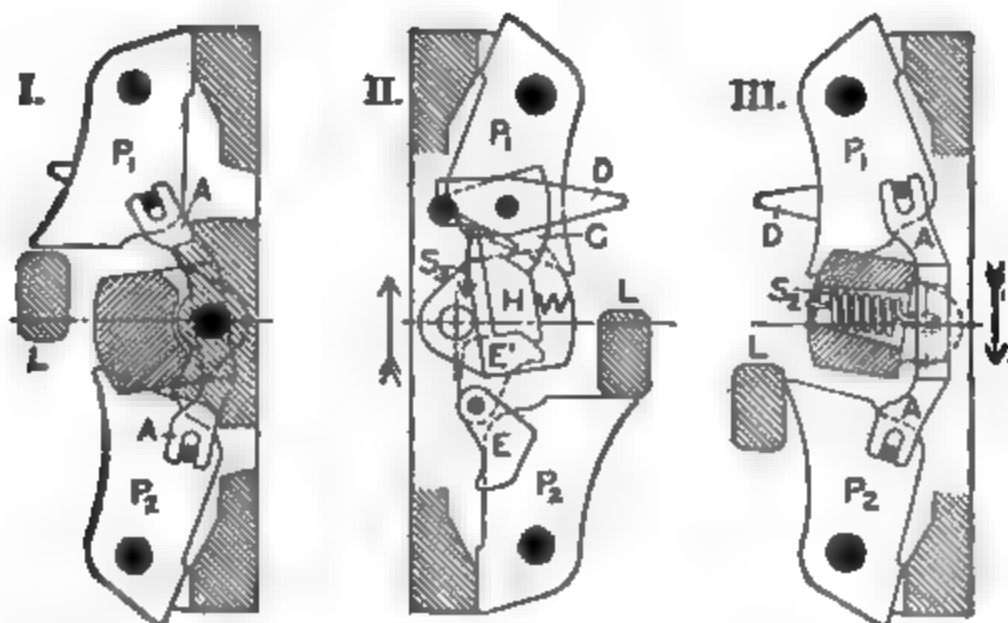


FIG. 568.—Operation of the Governor.

With or without these curves of dynamic action, it is easy to see that there will be a limiting speed up to which  $W$  will remain down, as in Fig. 568 I. If the speed rises above this limit,  $W$  will lift a little, come down on its stop with a knock. As the

## GOVERNORS OR RE

ed increases, so will the amplitud  
s far enough for the catch C to eng  
a and hold the weight up as at II.  
pawl P<sub>1</sub>, which raises L: but Fig. 5  
will be caught before the main lever  
so that P<sub>1</sub> will have to slide past the  
III., the arm-piece A, through which  
fast to W, but is held by a spring, be  
lugs—the action of this flexible conne

The weight W having been left t  
down-stroke, the lever L is raised d  
the piston; but as the governor desc  
up, releasing H from C, and letting W  
of the next down-stroke, P<sub>1</sub> will cat  
ordinary working, therefore, this ge  
from the engine for one revolution, or  
a multiple-expansion engine, however  
the same wide fluctuation in power-  
produced in a simple engine.

The emergency-stop device consist  
E, which is normally held down by th  
pushed out by a light spring at S<sub>1</sub>. I  
weight will be flung up from the full-li  
568 II.), propping up H, and staying  
hand.

During the few revolutions that  
after steam is shut off, the detent-arm  
both up and down; which is the rea  
instead of being fast to it. Note that  
to resist displacement of D in either di



# APPENDIX TO CHAPTER X.

## THE MOMENT OF CENTRIFUGAL FORCE.

The general case of a body rotating as in the fly-ball governor is set forth in Fig. 569, where  $OY$  is the axis of rotation and the body  $AB$  is capable of turning, in the plane of the drawing, about

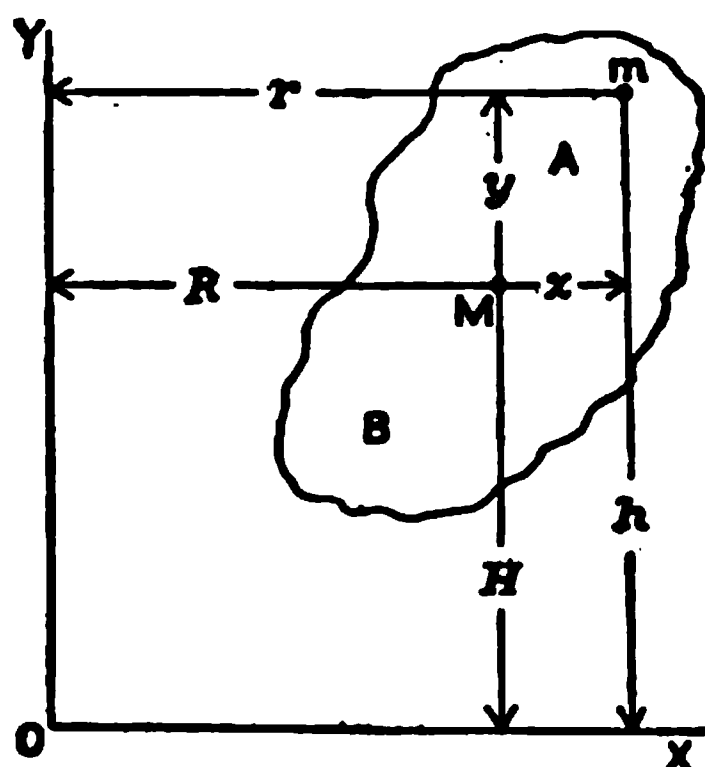


FIG. 569.—Moment of the Centrifugal force.

some point on the line  $OX$ . The co-ordinates  $R$  and  $H$  are respectively the rotation-radius and the moment-arm of the whole mass, at the center  $M$ . The centrifugal force of any particle  $m$  at radius  $r$  is

$$f = m\theta^2 r;$$

and its moment about any point on  $OX$  is

$$t = m\theta^2 rh.$$

Taking  $\theta=1$  for simplicity, we have that the whole body, of the mass  $M = \sum m$ , is subject to the moment

$$T = \sum t = \sum m r h.$$

From the figure

$$r = R + x, \quad h = H + y;$$

therefore

$$\begin{aligned} t &= m(R+x)(H+y); \\ T &= \sum mRH + \sum mxH + \\ &= MRH + \sum mxy; \end{aligned}$$

the summations  $\sum mx$  and  $\sum my$  both origin of  $x$  and  $y$  is at the center of

For the expression  $\sum mxy$  the title be fitting; and its determination is as the finding of "moment of inertia" cases are, First, any body symmetric co-ordinate axes, for which  $\sum mxy$  is slender bar, with its center-line passing on the rotation-axis, like the  $x=ay$  or  $\sum mxy=a \times \sum my^2$ .

General graphical methods, based of a particle having the constant of

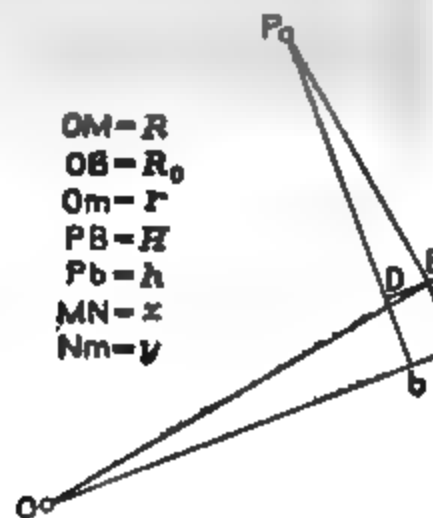


FIG. 570.—Moment of Centrifugal Force.

equilateral hyperbola referred to OX easily be developed; but lack of space of the subject forbid their presentation weight is very large in comparison with in Fig. 537, the error due to finding the for the real center of mass is insignificant belonging to the high-speed governor

is relatively small, the true moment was found to be about 33 per cent. greater than that got by concentrating the mass at the center of gravity. It appears, then, that in some cases neglect of this effect would introduce a considerable element of uncertainty into the design of the governor.

It is of interest to prove that in the shaft-governor no such secondary effect exists as that which has just been discussed. In Fig. 570, O is the center of rotation and P the pivot-point for an arm whose center of mass is at M. For a particle at m the moment about P due to centrifugal force is

$$t = m\theta^2 r h.$$

**The useful geometrical relation is**

$$h = PD + Db = H \frac{R+x}{r} + R_s \frac{y}{r},$$

determined by the similarity of the triangles  $O_mN$ ,  $PBD$ , and  $OBE$ , right-angled at  $N$ ,  $D$ , and  $E$  respectively. Then, for  $\theta=1$ ,

$$t = m\tau h = m[H(R+x) + R_y];$$

by summation this gives

$$\begin{aligned} T &= \Sigma t = MRH + \Sigma m_x H + \Sigma m_y R, \\ &= MRH. \end{aligned} \quad (366)$$

## CHAPTER STEAM-ACTION IN THE MULTIPLE

### § 66. Simplest Conditions: the Re

(a) GENERAL CONDITIONS.—The discussion of steam-action are developed general ideas about compound engines § 22 (b) and (c); and we are now to the synthetic side of the matter for which is given in § 23. At first considering the possible form, we shall neglect all the major effects alone by making the same as those which determine the situation and 29. These conditions are:

1. There are no losses of pressure or valve-action, so that the direct cut-off, release, etc.

2. All operations of expansion are mixing of different bodies of steam  $pv = C$ .

3. Only the action of the working account. This requires the assumption of no clearance-volumes, and that the sensible fluctuation in pressure will be inflow and outflow of steam.

The whole discussion to follow will be for a two-stage or "compound" engine: the method for this type can easily be extended to other arrangements.

As to the general manner of treatment, it must be understood that the mathematical method has a very limited field of usefulness in connection with this subject, serving effectively only for the deduction of a few fundamental relations between the principal quantities involved in the working in the engine. On the least departure from the primary, ideal case just outlined, the graphical method becomes the only one that is practically useful. And back of any discussion along the lines of simple theory lie a number of secondary actions, whose effect and magnitude can be found or estimated only by experience—that is, by the study of indicator diagrams from actual engines.

(b) THE ENGINE WITH INFINITE RECEIVER.—This heading expresses with mathematical exactness the idea of receiver-effect just set forth under condition 3. Practically, it means that the pressure in the receiver, which determines the common line of exhaust from the high-pressure cylinder and of admission to the low-pressure, is to remain constant so long as the total rate of flow through the engine is constant. Then the ideal diagrams (which more or less closely circumscribe those actually drawn by the indicator) take the form shown in Figs. 574 and 575, where ABCHJ and JDEFG coincide along the horizontal line JH. In both these figures the heavy-line diagrams marked by the subscript 2 are supposed to correspond with the best working of the engine, at its proper, rated load.

Under the assumption that  $pv = C$ , and since the same quantity of steam passes through both cylinders, we have that the individual expansion-curves BC and DE must be parts of one continuous equilateral hyperbola, with its origin at M. Usually, however, the total expansion is broken, by the drop from C to H, with a loss from the disposable work of an amount represented by the area CHD. This receiver-drop is analogous to that at the final release; and the argument against complete expansion, given in § 17 (d), applies with equal force to the high-pressure cylinder. But while, for the best results in the mechanical sense, CH ought not to be less than the reduced mean friction of the high-pressure engine, an exaggerated drop may cause a great loss in efficiency—as is exemplified by the diagrams marked 3 and 4 on Fig. 574.

on the other hand, note how, with expansion-curve  $B_1C_1$  drops below

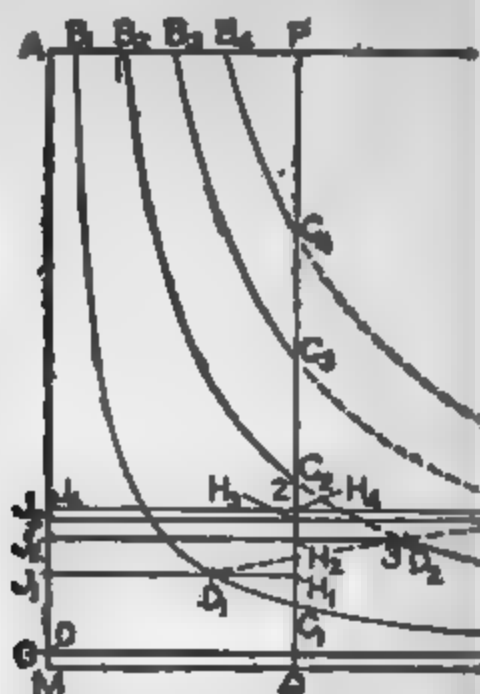


FIG. 574.—Diagrams Showing Equal Division of Work between two cylinders with equal cut-off in both cylinders.

the negative-work loop  $D_1C_1H_1$ , just similar conditions, and with a corresponding variation in the cut-off in the L.P. cylinder.

(c) DIVISION OF WORK BETWEEN TWO CYLINDERS.—If the quantity of steam admitted to the engine is fixed, and the H.P. cut-off, the receiver-pressure and the receiver-volume are determined by the conditions of the L.P. cylinder, the proportion of work done in the L.P. cylinder can be made earlier, and vice versa, as shown in the diagrams. That is, variation in a cut-off in the H.P. cylinder has the effect to variation in a cut-off which determines the quantity of steam admitted.

A requirement very often imposed is that the total work done shall be equal to the work done in two cylinders. This requirement is not shown in the cases shown on Fig. 574, each area  $A_1$  is equal to the corresponding  $JDEFG$  (coincident

is interesting to note how, under this condition, the L.P. cut-off varies with the H.P. Here D has a somewhat wider range over the stroke-line MN than has B on AP, and D is always later than B. It is evident that the exact relation between these two cut-offs must be largely dependent upon the cylinder-ratio.

Fig. 574, with coincident change in both cut-offs, stands for one type of relative valve-action; the other type is illustrated by Fig. 575, where the L.P. cut-off is kept constant. With equal

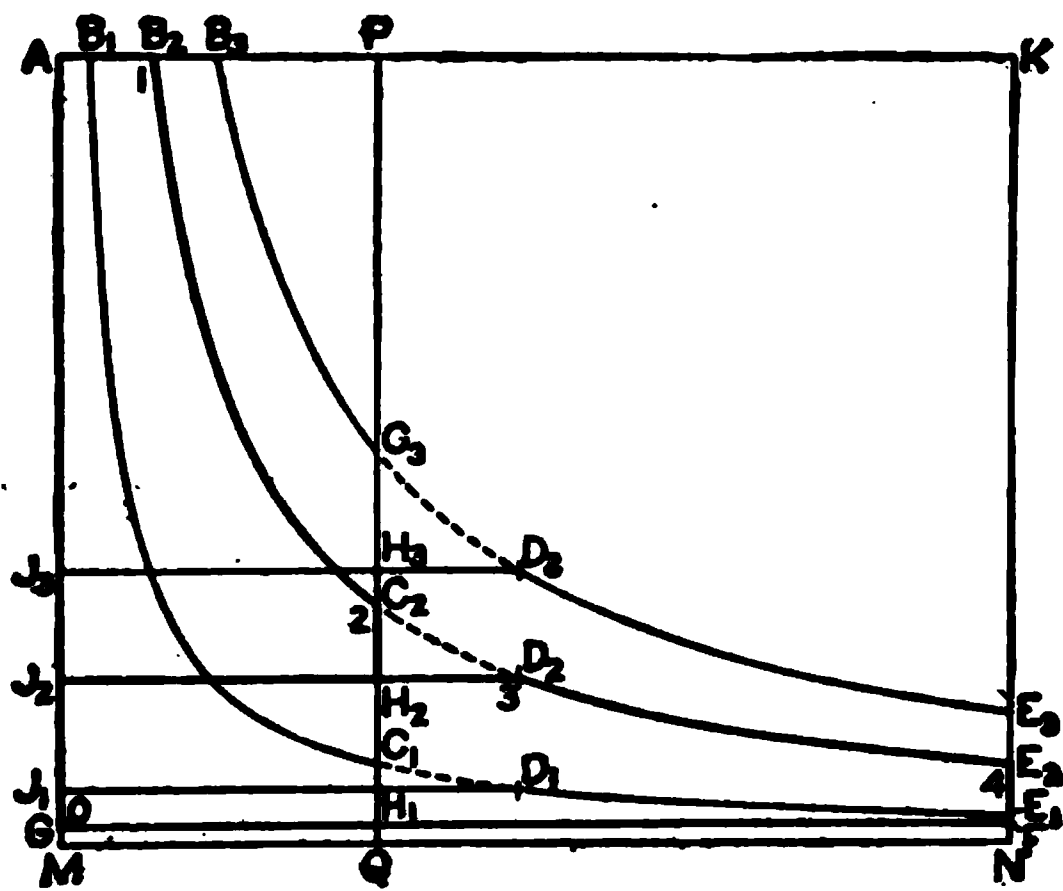


FIG. 575.—Constant Cut-off in the Low-pressure Cylinder.

division for the middle curve, it is made apparent that the H.P. cylinder takes the larger share of a smaller load, the smaller share of an increased load. Over against this variation in division is the advantage that the receiver-drop loss does not rapidly increase with the load as it does in Fig. 574.

(d) THE CASE OF COMPLETE EXPANSION.—The simplest relations as to work-division exist under the conditions represented by Fig. 576 I. There is complete expansion in each cylinder, including the last, and (for a three-stage engine) the total area ABCD is divided into three equal parts by the horizontal lines EF and GH. Following § 17 (e) and using the subscripts 1, 2, and 4 to designate pressures and volumes at the four

critical points B, E, G, and C, we have

$$\begin{aligned} \text{Area ABEF or } A_1 &= p_1 v_1 \left( 1 + \log_e \frac{v_2}{v_1} \right) - p_2 v_2 \\ &= p v \log_e r_1, \quad \dots \dots \dots (37) \end{aligned}$$

letting  $\mu$  stand as a general value for the curve BC. Similar:

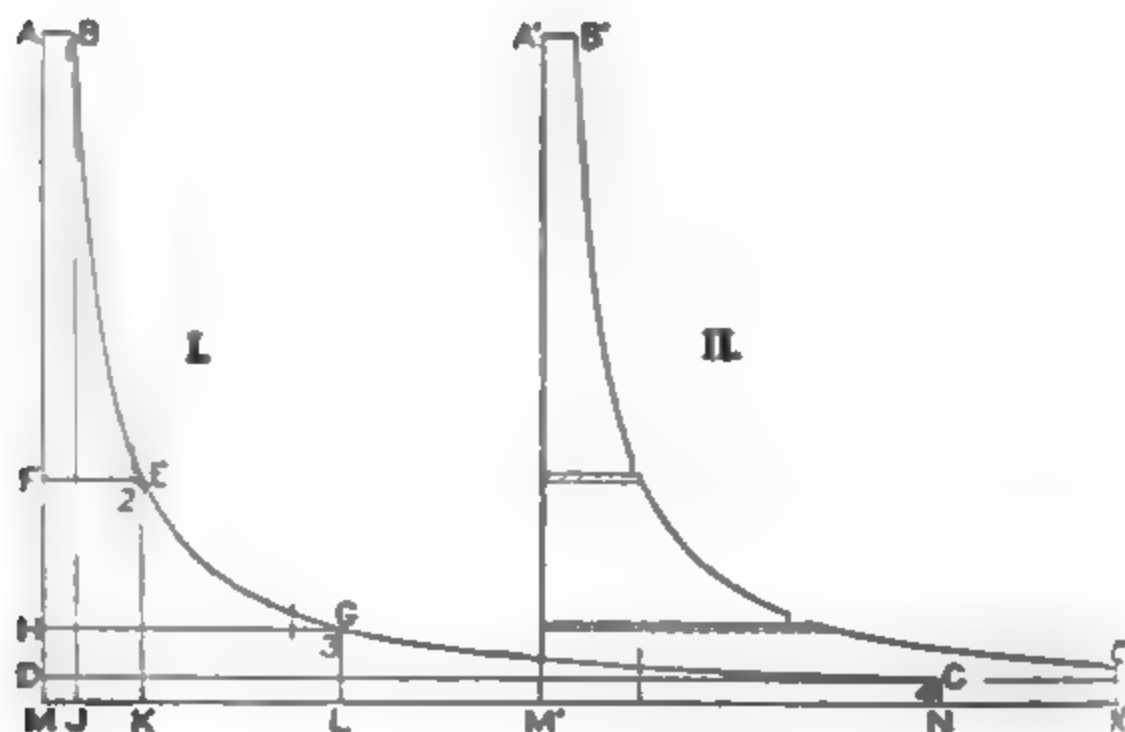
$$\text{FECH} = A_2 = pr \log_e r_2; \quad \text{HGCD} = A_2 = pv \log_e r_2$$

For equal partial areas  $A_1$ ,  $A_2$ , and  $A_3$ , the individual expansion ratios must be equal, or

$$\frac{r_1}{t_1} = \frac{r_2}{t_2} = \frac{r_3}{t_3} = r;$$

further, the total ratio  $R$  or  $r_1, r_2$  will evidently be the product of  $r_1, r_2$ , and  $r_3$ ; wherefore, in this case,  $R=r^3$ ; or in general for  $n$  stages,

$$R=r^2, \quad (37)$$



**FIG. 576.**—Complete Expansion and the Introduction of Terminal Drop

With complete expansion therefore, or under the condition that the total volume of is equal to the cut-off volume of the next lower cylinder each cylinder



were identical also with the ratio of successive cylinders, will be the  $n$ th root of the total ratio of expansion. In Fig. 576, for instance, the value of  $r$  is 3, and that of  $R$  or  $DC/AB$  is 27.

The first modification from this wholly ideal case is shown at II. in Fig. 576. The expansions are cut short at the volumes indicated by dotted lines on I.; and to equalize areas the cut-offs are moved up a little in stages 2 and 3, raising the receiver-pressures above the original levels, which are dotted in on II. Usually, the amount cut from the last stage is relatively much greater than from the earlier stages.

(e) MATHEMATICAL RELATIONS for the compound engine, as represented by Figs. 574 and 575, will now be deduced—using the subscripts as indicated on the figures, with 0 for the exhaust-pressure. The fundamental equation, embodying the assumption that the condition  $pv=C$  is maintained throughout the operation, is

$$p_1v_1 = p_2v_2 = p_3v_3 = p_4v_4. \quad . \quad . \quad . \quad . \quad . \quad (373)$$

For the areas we have

$$ABCHJ = A_1 = pv \left( 1 + \log_e \frac{v_2}{v_1} \right) - p_3v_2; \quad . \quad . \quad . \quad (374)$$

$$JDEFG = A_2 = pv \left( 1 + \log_e \frac{v_4}{v_3} \right) - p_0v_4. \quad . \quad . \quad . \quad (375)$$

Equating and dividing by  $pv$ , using the particular values in (373) as convenient, we get

$$\log_e \frac{v_2}{v_1} - \frac{v_2}{v_3} = \log_e \frac{v_4}{v_3} - \frac{p_0}{p_4}. \quad . \quad . \quad . \quad . \quad . \quad (376)$$

If there is to be no receiver-drop, or if  $v_3 = v_2$ , and if further we let  $v_1 = Ev_4$ , this becomes

$$\log_e \frac{v_2}{Ev_4} - \log_e \frac{v_4}{v_2} = 1 - \frac{p_0}{p_4},$$

or

$$\log_e \frac{1}{E} \left( \frac{v_2}{v_4} \right)^2 = 1 - \frac{p_0}{p_4};$$

whence

$$\frac{v_2}{v_1} = \sqrt{E} \sqrt{\text{anti-log} \left( 1 - \frac{p_2}{p_1} \right)} = C \sqrt{E}. \quad (377)$$

This  $E$  is the reciprocal of  $R$  above; the cylinder-ratio  $v_1/v_2$  is no longer the square root of  $R$ , but it is proportional to that root according to a factor which depends upon the ratio of final pressure-drop,  $p_2/p_1$ .

If instead of making  $v_1 = v_2$ , we allow a certain proportion of receiver-drop by letting  $v_2 = \pi v_1$  (which makes  $p_2 = \pi p_1$ ,  $\pi$  being greater than one), the formula becomes

$$\log_e \frac{\pi}{E} \left( \frac{v_2}{v_1} \right)^2 = \frac{1}{\pi} - \frac{p_2}{p_1},$$

$$\frac{v_2}{v_1} = \sqrt{\frac{E}{\pi}} \sqrt{\text{anti-log} \left( \frac{1}{\pi} - \frac{p_2}{p_1} \right)}. \quad (378)$$

**EXAMPLE 1.** Let  $p_1 = 100$  lbs. and  $p_2 = 2.5$  lbs.; take  $v_1$  to be 1.00 for  $p_1 = 10$  lbs. or  $v_1 = 0.1$  and  $\pi = 1.25$ , find  $v_2$ , or the size of the H.P. cylinder for equal work-division.

Here

$$\frac{1}{\pi} - \frac{p_2}{p_1} = 0.80 - 0.25 = 0.55.$$

To get the common log, divide by 2.3026, which gives

$$0.55 \div 2.3026 = 0.23886.$$

Then, with  $E = 0.1$ ,

$$\frac{v_2}{v_1} = \sqrt{\frac{0.1}{1.25}} \sqrt{1.733} = \sqrt{0.1386} = 0.3723;$$

and  $v_2 = 1.25 v_1 = 0.4654$ .

Without receiver-drop, or with  $v_1 = v_2$ , we should get from (377),

$$\frac{v_2}{v_1} = \sqrt{0.1 \times 2.117} = 0.4592.$$

In Figs. 574 and 575 the overall proportions are the same as in the above example, and the cylinder-ratio  $v_1/v_2 = 0.332$ .

For Case 2 this makes the drop about 31 per cent. (of  $p_2$ ), or gives  $n$  the value  $1/0.69=1.45$ , and fixes  $v_2/v_1$  at 0.48 for equal work-division. Comparing this 0.48 with the values 0.465 and 0.459 above, we see the important fact—evident also from a mere inspection of the diagrams—that for a given admission-volume  $v_1$  in the first cylinder, there can be quite a wide variation in the cylinder-volume  $v_2$ , with only a small effect upon either the total work or the L.P. cut-off  $v_3$ , which will keep the division equal. Until the receiver-drop becomes quite large, it is the only element that is much affected by thus varying  $v_2$ .

We have therefore the important practical conclusion (from the whole discussion up to this point), that the principal element in determining work-division is the receiver-pressure. As the first step in the layout of preliminary diagrams for a proposed engine which is to have reasonably full expansion in the upper stage or stages, this pressure (or the corresponding volume) can be closely enough calculated by the method of Eq. (377): then the succeeding steps, in the direction of closer approximation of the diagram toward the actual form, are matters for graphical rather than mathematical determination.

As an instance of the limited utility of the mathematical method, we may consider the problem involved in drawing Fig. 574, where, with an assumed cylinder-ratio, the two areas are to be made equal. To find  $v_3$ , we should put Eq. (376) into the form

$$\log_* \frac{v_1}{v_3} + \frac{v_2}{v_3} = \log_* \frac{v_2}{v_1} + \frac{p_2}{p_1}, \quad . \quad . \quad . \quad . \quad (379)$$

collecting the known terms into the right-hand member: With  $v_1$  thus involved, the equation can be solved only by a trial method, which is at least as much trouble as to find the exact location of the division-line JD by means of successive measurements with the planimeter.

(f) THE GENERAL CASE OF WORK-DIVISION.—One more mathematical deduction finds place properly at this point, to cover the case presented by Fig. 577. Here the area  $A_1$  is not equal to  $A_2$ , but bears to it the definite relation expressed by the factor  $q$  in



the equation

$$p_1 r_1 \left( 1 - \log_e \frac{r_2}{r_1} \right) - p_2 r_2 = c \left[ p_2 r_2 \left( 1 + \log_e \frac{r_1}{r_2} \right) - p_0 r_0 \right] \quad (372)$$

Very often the average value of the steam-measure  $pr$  is not the same in the lower stage as in the higher—in an unjacketed engine it is usually less, because of increased cylinder-condensation. To represent such a condition we change Eq. (373) to

$$p_1 r_1 = p_2 r_2 = m p_3 r_3 = m p_0 r_0 \quad \dots \quad (373)$$

Now defining the receiver-drop as for Eq. (378) by the relation

$$p_2 = n p_3 \quad \text{whence} \quad r_3 = \frac{n}{m} r_2$$

and again letting  $r_1 = E r_0$ , we get from (380), through division by  $p_2 r_2$ ,  $p_1 r_0$ , or  $p_1 r_1$ , the equation

$$m \left( 1 - \log_e \frac{1}{E} \frac{r_2}{r_0} \right) - \frac{r_2}{r_0} = q \left( 1 + \log_e \frac{m}{n} \frac{r_2}{r_3} - \frac{p_0}{p_2} \right).$$

This transforms to

$$m + \log_e \left( \frac{1}{E} \right)^m \left( \frac{r_2}{r_0} \right)^m - \frac{m}{n} = q + \log_e \left( \frac{m}{n} \right)^q \left( \frac{r_2}{r_3} \right)^q - q \frac{p_0}{p_2},$$

or

$$\log_e \left( \frac{1}{E} \right)^m \left( \frac{n}{m} \right)^q \left( \frac{r_2}{r_0} \right)^{m+q} = q - m + \frac{m}{n} - q \frac{p_0}{p_2} \quad \dots \quad (382)$$

**EXAMPLE 2.** In Fig. 577,  $A_1 = 40$  per cent.,  $A_2 = 60$  per cent., of the total effective area, or  $q = \frac{2}{3}$ ; the receiver-drop is one-sixth, or  $n = \frac{5}{6}$ ; of the product  $pr$ , 10 per cent. is lost in passing to the lower stage, or  $m = \frac{4}{5}$ ; the ratio of initial cut-off is  $E = \frac{1}{10}$ ; and the final pressures are  $p_1 = 8$  lbs.,  $p_2 = 2$  lbs., which makes the initial pressure  $8 \times 16 \times \frac{1}{2} = 142.2$  lbs.

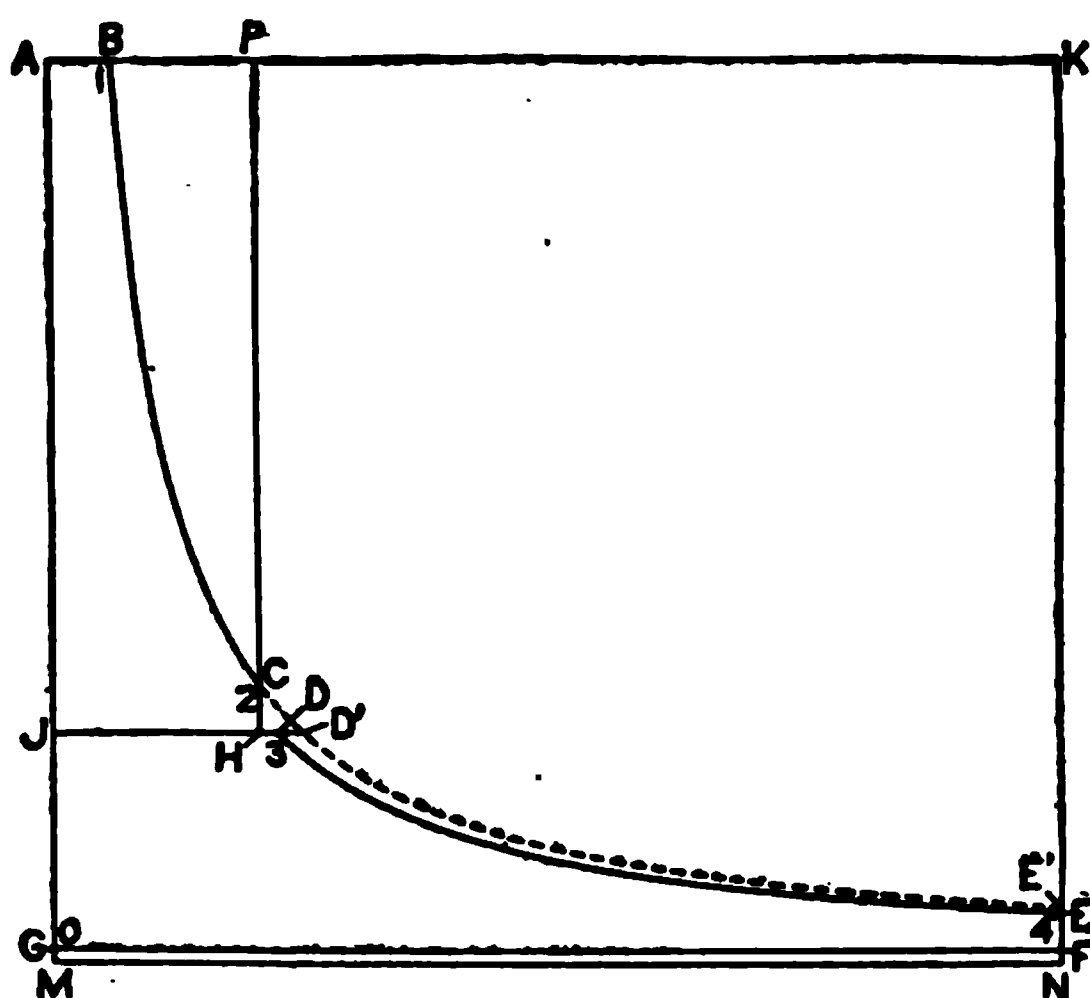
**Using Eq. (382) with these data, we get**

$$\log_e 16^{\frac{1}{2}} 1.08^{\frac{1}{3}} \left( \frac{v_2}{v_1} \right)^{\frac{1}{6}} = -0.6667 - 1.1111 + 0.9259 - 0.1667$$

$$= -0.3148.$$

$$\frac{v_1}{v_4} = \left( \frac{1.3700}{21.77 \times 1.0524} \right)^{\frac{1}{1.4}} = 0.2050.$$

**Fig. 577 is laid off with this ratio of AP to AK, and the measured areas check up to the assumed value of  $q$ . For a purely graphical determin-**



**FIG. 577.—A General Problem in Work-division.**

ation, we should have to note that as CH moves sidewise, H will travel along a hyperbola below BC, located by the ratio  $n$ : then it would be easy to draw coincident CH and JD lines and measure areas, until the desired proportion was secured.

As to the division of work in the above example, it is such as might be desired in a three-cylinder compound engine—giving the H.P. cylinder 40 per cent. and each of the L.P.'s 30 per cent. of the total work. and the whole method would apply equally to

the problem of dividing up the total expansion in a three- or four-stage engine.

(g) GENERAL CONCLUSIONS.—Some of the following statements are repetitions, all of them are quite evident conclusions from the diagrams and formulas that have been presented—chiefly from the diagrams.

A. The fundamental data for any steam-operation, the leading dimensions of the cycle as a whole, are

- $p_1$  = admission-pressure;
- $p_4$  = pressure at end of expansion;
- $p_0$  = exhaust-pressure;
- $v_4/v_1$  = total ratio of expansion.

To find whether  $p_4v_4 = p_1v_1$ , or, more generally, how the steam-measure  $pv$  in the L.P. cylinder compares with the  $pv$  in the H.P. cylinder, is one of the principal objects in combining actual indicator diagrams—as pointed out in § 23.

B. In a compound engine with a given cylinder-ratio, the variable elements of the cycle are

1. The load on the engine;
2. The total ratio of expansion, as fixed by the H.P. cut-off.
3. The L.P. cut-off;
4. The exhaust-pressure.

C. With the load constant, the work done in the H.P. cylinder is increased

By making the L.P. cut-off later;

By raising the boiler-pressure: this slowly decreases  $pv$  or the amount of steam admitted to perform a given total amount of work per revolution; but it adds to the H.P. diagram, by raising the admission-line and lowering the exhaust-line, more rapidly than it subtracts by drawing inward the expansion-curve:

By raising the exhaust-pressure: this subtracts more from the bottom of the L.P. diagram than is added on account of the increase in  $pv$  and the corresponding rise in the receiver-pressure.

D. In a given engine, the preservation of an equal division of load requires that the L.P. cut-off will vary with the H.P., and in the same direction. cut-off constant,

creasing the load increases the share taken by the L.P. cylinder.

E. In proportioning an engine, with a view to a certain division of power and a reasonable relative amount of loss by receiver-drop,

The H.P. cylinder must be relatively smaller as the total ratio of expansion is greater:

With a fixed L.P. cut-off and with a constant amount of steam admitted to the H.P. cylinder per revolution, the volume of the H.P. cylinder can be varied over quite a range with comparatively little effect upon the division of work, but with a great influence upon the receiver-drop—compare (e), under Example 1:

If the L.P. cut-off is varied in a proper \* relation to the size of the H.P. cylinder, making the latter larger gives it a larger share in the power developed.

### § 67. The Direct-expansion Engine.

(a) THE ENGINE WITH NO RECEIVER.—In approaching the actual engine, from the ideal case just discussed, we begin with the type farthest from the engine with “infinite” receiver, and take up first that which has no receiver at all—for one strong reason, because the relations as to volumes are so much simpler when the strokes are simultaneous, and this is the simplest case under that type of arrangement. Still assuming that the clearance-volume is zero and the valve-action ideal, we get the operation represented in Fig. 578. When the high-pressure expansion ends, at C, communication is opened to the large cylinder; and during the whole of the return stroke the intermediate valve must be kept open—see the arrangement in Fig. 269. Since the L.P. piston is so much the larger, there is expansion, from  $v_2$  at the beginning to  $v_4$  at the end of this return stroke, with a drop in pressure shown, for the H.P. cylinder, by the curve CE'. The total volume increases just as if a single piston of the area  $(A_2 - A_1)$  started from a position where the volume back of it was  $v_2$  and travelled through a distance equal to the stroke of the engine—this being the result—

---

\* That is, so as to keep the proportion of receiver-drop about constant.

it of a loss due to the return of the gain due to the advance of the L.L.

There are several ways of representing a low-pressure cylinder. Laid off on the same diagram will fit under the H.P. diagram would be drawn, for instance, if the

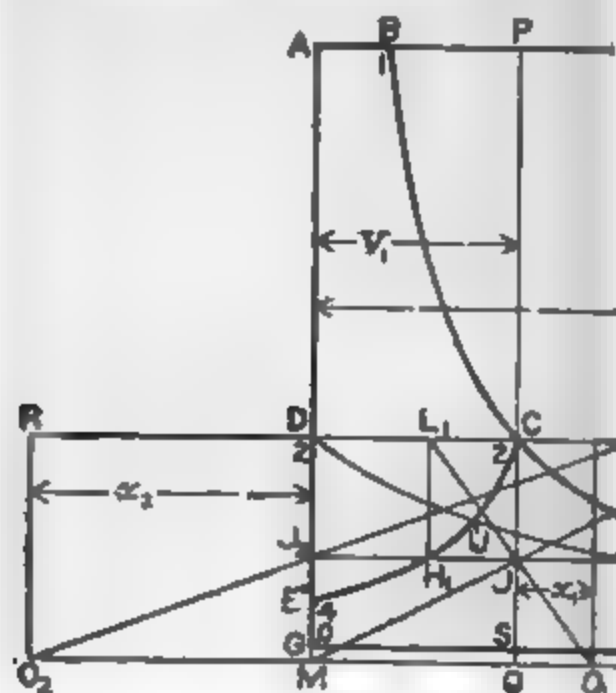


FIG. 578.—The Engine with

to both cylinders by suitable piping taken with the same spring and on a common scale of volume, so as to relative proportion, this diagram becomes lap DE'U just equal to the apparent clearly the continuity of the expansion to the equivalent form E'CEFG: the whole area into respective parts, of receiver-pressure in the preceding QJ, the total steam-volume is J<sub>2</sub>H; the small cylinder, then that in the large be the same as H<sub>1</sub>H.

Since change of volume during the process is proportional to the travel of either piston, the expansion curves in both cylinders are simple hyperbolas, which can be drawn



origin, in the usual manner. To find the origins, we note that either base-line, QM or MN, now represents a volume-change of the amount  $(V_2 - V_1)$ : then for  $O_1$  we make

$$QO_1 = x_1 = QM \frac{V_1}{V_2 - V_1}, \quad \dots \quad (383)$$

and for  $O_2$

$$MO_2 = x_2 = MN \frac{V_1}{V_2 - V_1}. \quad \dots \quad (384)$$

(b) MATHEMATICAL RELATIONS.—The division of area between the two partial diagrams can be most easily calculated as follows:

For the whole figure ABCEFG, the mean effective pressure is

$$P_1 = \frac{p_1 v_1 \left(1 + \log_e \frac{v_4}{v_1}\right)}{v_4} - p_0. \quad \dots \quad (385)$$

And for the low-pressure diagram DEFG, on the base MN, the M.E.P. is the same as that of the figure CEFS on the base QN, or is

$$P_2 = \frac{p_2 v_2 \log_e \frac{v_4}{v_2}}{v_4 - v_2} - p_0. \quad \dots \quad (386)$$

For equal work-division,  $2P_2 = P_1$ , or

$$2p_2 \frac{v_4}{v_4 - v_2} \log_e \frac{v_4}{v_2} - 2p_0 = p_1 \left(1 + \log_e \frac{v_4}{v_1}\right) - p_0;$$

whence

$$\frac{v_4}{v_4 - v_2} \log_e \frac{v_4}{v_2} = \frac{1}{2} \left(1 + \log_e \frac{v_4}{v_1} + \frac{p_0}{p_2}\right). \quad \dots \quad (387)$$

EXAMPLE 1. In Fig. 578,  $v_4 = 1.00$ ,  $v_1 = 0.10$ ,  $p_1 = 10$ ,  $p_0 = 2.5$ . To find  $v_2$  we have

$$\frac{v_4}{v_4 - v_2} \log_e \frac{v_4}{v_2} = \frac{1 + 2.3026 + 0.25}{2} = 1.7763.$$

For  $v_2 = 0.3$ ,  $1/0.7 \times \log 3.333 = 1.429 \times 1.2040 = 1.720$ ;

For  $v_2 = 0.25$ ,  $1/0.75 \times \log 4.000 = 1.333 \times 1.3863 = 1.848$ ;

For  $v_2 = 0.28$ ,  $1/0.72 \times \log 3.571 = 1.389 \times 1.2729 = 1.768$ ;

For  $v_2 = 0.275$ ,  $1/0.725 \times \log 3.636 = 1.379 \times 1.2909 = 1.780$ .

Evidently, 0.277 would be the cor

Graphically, it would be a simple  
positions until the mean height of CE  
equal half the mean height of ABEFG

(c) IDEAL ACTION WITH REC  
properly the next in order, is illus  
assumed proportions, three different  
In discussing this or any similar ca  
 $p$  and  $v$  with subscripts referring t  
for the fixed volumes we use cap  
designate quantities belonging to

$V_1$  = volume of high-p

$V_2$  = volume of low-pr

$R$  = volume of receive

For the important ratios the sy

$e_1 = v_1/v_2$  = H.P. ca

$e_2 = v_{4''}/v_7$  = L.P. c

$E = v_1/v_7$  = total r

$n = V_1/V_2$  = cylind

$a = R/V_1$  = receive

Now in the action represented  
the steam from cylinder 1 exhaust  
it meets and mixes with a body of  
the resulting pressure  $p_3$  begins a c  
3'4' on the H.P. diagram, 3''4'' on  
the cut-off at 4. here the whole bo  
parts, and then the working-steam  
the receiver-steam is compressed by  
We wish to be able to determine th  
of the important, numbered points, i  
curves.

First, write expressions for all th

quantities; in any actual case, it is well to reduce all the volumes to numerical values, in terms of either  $V_1$  or  $V_2$ , as a base-unit.

$$\begin{aligned} v_1 &= e_1 V_1 = E V_2; & v_2 &= V_1; \\ v_3 &= V_1 + R; & v_5 &= R; & v_7 &= V_2; \\ v_4 &= [(1 - e_2) V_1 + R] + e_2 V_2 = v_{4'} + v_{4''}. \end{aligned}$$

If necessary, make a sketch diagram of the cylinders with the pistons in positions 3, 4, and 5, to see that these expressions are

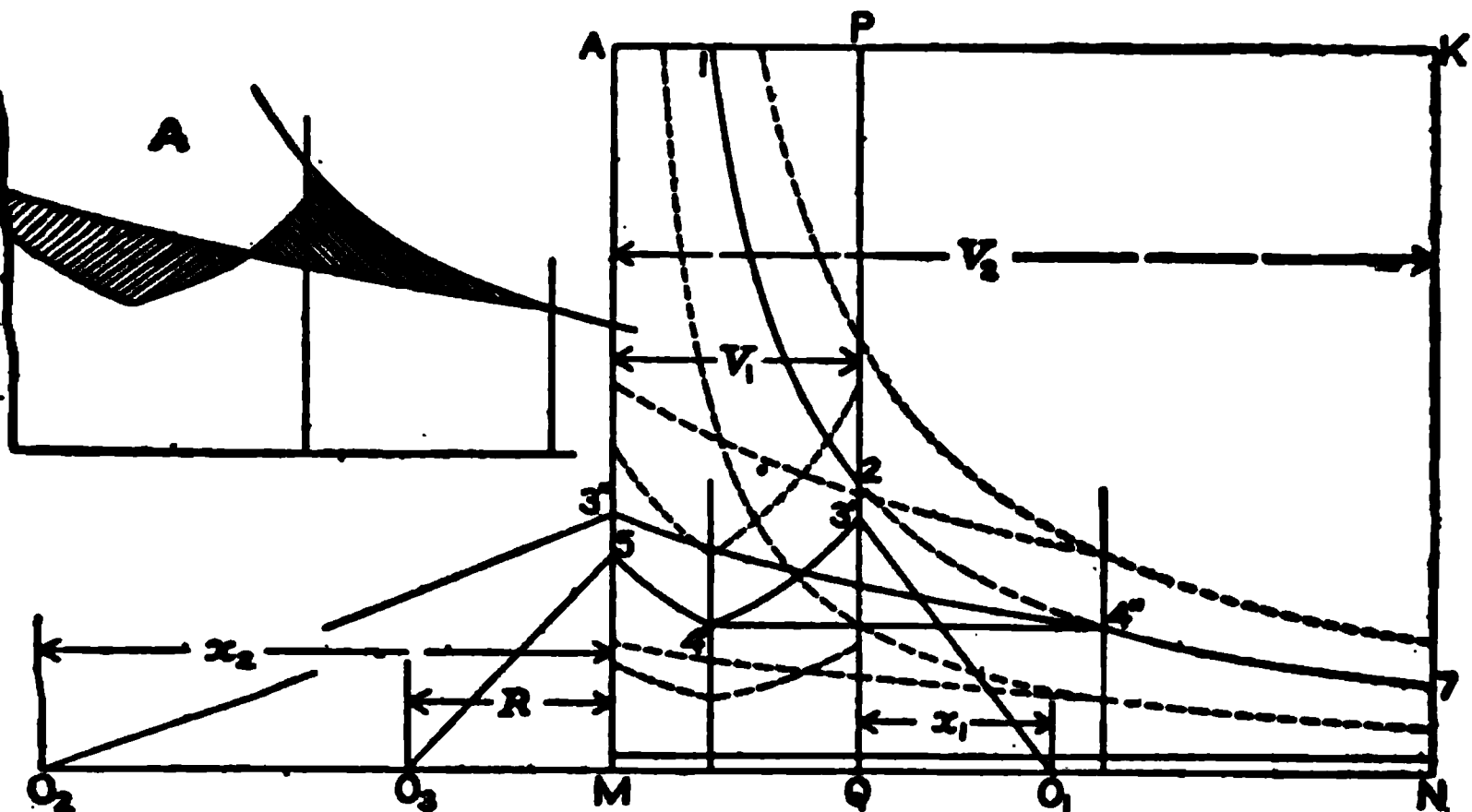


FIG. 579.—Direct-expansion Engine with a Receiver.

correct. At 4 (low-pressure cut-off) the total volume  $v_4$  is divided into two parts:  $v_{4''}$ , all in the L.P. cylinder, contains the working-steam;  $v_{4'}$ , in receiver and H.P. cylinder, contains the receiver-steam.

As in the preceding discussions, the fundamental relation is

$$p_1 v_1 = p_2 v_2 = p_4 v_4 = p_7 v_7. \quad . \quad . \quad . \quad . \quad . \quad (388)$$

This determines the pressure  $p_4$ ; and with that known, we easily calculate  $p_3$  and  $p_5$  from the equations  $p_3 v_3 = p_4 v_4$  (for the combined bodies of steam) and  $p_5 v_5 = p_4 v_{4'}$  (for the receiver-steam alone). Incidentally,  $p_3$  could be found, after  $p_5$  was known, by the relation

$$p_3 v_3 = p_2 v_2 + p_5 v_5. \quad . \quad . \quad . \quad . \quad . \quad (389)$$

## 1 THE MULTIPLE-EXPA

though it is not here necessary to m  
ls operation of mixing.

EXAMPLE 2.—In Fig. 579, taking 1  
and for the full-line diagram,  $e_1 = 0.4$ ,  $e_2$   
12; and the several values work out as

$$\begin{aligned} v_1 &= 0.12, & p_1 &= 100; \\ v_2 &= 0.30, & p_2 &= 40; \end{aligned}$$

$$v_3 = (0.4 \times 0.3 + 0.25) + 0.6 = 0.97;$$

$$p_3 = 20 \times 0.97 \div 0.4$$

$$v_4 = 0.37;$$

$$p_4 = 20 \times 0.37 \div 0.2$$

By mixing,

$$p_5 \times 0.55 = 40 \times 0.30 + 20$$

$$p_5 = 35.2.$$

Graphically, we get the point 4'' by lo  
on MN according to  $e$ ; then locate the  
and project over the pressure at 4'' to  
and 4'3' can be drawn from fixed pole:  
 $(1 - e_2)V_1 + R + e_2V_2$ , equal to  $V_1 + R +$   
and since the full stroke-line MN or QM  
 $(V_2 - V_1)$ , we make

$$\frac{x_1}{QM} = \frac{x_2}{MN} = \frac{V_1}{V_2}$$

For the curve 4'5 the pole  $O$  is got by  
to the regular scale, as indicated on the fig

(d) DIVISION OF WORK.—Even wit  
involved in Fig. 579, it becomes pract  
a general relation between the areas of  
tive cylinders. It would be better,  
rather than "impossible", as appears  
of relation for equal areas in a diagra

tion, whose derivation would here take up too much space, is

$$\frac{1+n}{1-n} \frac{(1-e_2+a)n+e_2}{e_2} \log_e \frac{(1-e_2+a)n+e_2}{n(1+a)} + n \frac{1-e_2+a}{e_2} \log_e \frac{1-e_2+a}{a} \\ - \log_e \frac{n}{E} = 1 - \log_e \frac{1}{e_2} + \frac{p_0}{Ep_1}. \quad . \quad . \quad . \quad . \quad . \quad (389)$$

All the symbols have already been defined in the last article. It would be easier to cut and try and measure areas a good many times, than to attempt to satisfy such an equation.

For this necessary cut-and-try graphical method, the starting point would be furnished by Fig. 578 or Eq. (387)—compare the actual diagrams in Figs. 583, 597, and 598. As the receiver is made larger, however, and with the effect of the valves and steam-passages in retarding fluctuations of pressure, the latter become smaller, and the engine approaches the action belonging to the case discussed in § 66, Fig. 604 being an apt example.

The way to measure the loss by receiver-drop on Fig. 579 is to compare the two areas, of overlap and of deficiency, shaded at A. Since the receiver-volume enters into the common expansion, the L.P. diagram cannot be closely fitted in against the H.P., after the manner of E'CEFG in Fig. 578. With the small drop shown on Fig. 579 the loss of area is insignificant.

(e) THEORETICAL DIAGRAMS FOR AN ACTUAL ENGINE.—In a real engine, the actions which have just been discussed are further complicated by the cylinder-clearances, with the consequent presence of a body of clearance-steam in each cylinder. To develop a solution of the problem thus presented, we will now analyze an actual design, choosing one in which the secondary effects are large. An excellent example for this purpose is supplied by the locomotive illustrated in Figs. 256 to 258 (compare also Fig. 437); and the proportions used in Fig. 580 are taken from a pamphlet on the steam-distribution in this engine and from other published descriptions.

The cylinders are 15" and 25" by 26"; and the several volumes involved, expressed, not in cubic measure, but in terms of either nominal cylinder as unity, are



giving the bases  $M_1N_1$ ,  $M_2N_2$ , and to put in the lines of admission-pressure and of exhaust-pressure. The former is taken at 185 lbs. absolute, which might be realized from a boiler-pressure of 180 lbs. by gage (depending upon speed and opening of throttle-valve); the exhaust is at 18 lbs., a rather low value for a locomotive.

The primary variable is the high-pressure cut-off, here made effective at half-stroke; that is, the ideal expansion-curve starts at 2, half-way from 1 to Q; and then the estimated admission-curve, sketched in, locates the actual, mechanical cut-off at 3. Having extended the expansion hyperbola well down into the low-pressure space, we next draw the valve-diagram that will give the cut-off at 3, and determine all the other events of the valve-action. In this diagram, the letters A, C, R, and K designate admission, cut-off, release, and compression, with sub one for the H.P. and sub two for the L.P. cylinder. Disregarding connecting-rod effect, we transfer these determinations to the respective stroke-lines  $M_2N_2$ ,  $M_1N_1$ , marking the ordinates with the same designating letters. For convenience, the diameter MN is made half of  $M_2N_2$ ; then any event is located on  $M_2N_2$  by doubling the horizontal distance given by the valve-diagram; and for  $M_1N_1$  we get a simple reduction-diagram by making  $N_2B$  equal  $M_1N_1$  and drawing  $M_2B$ .

As soon as we reach the H.P. release at  $R_1$  or point 4, we encounter the fact that the steam in the cylinder is now to mix with a body of receiver-steam of unknown pressure, that is, of unknown quantity. To get a known condition, we drop to the exhaust operation, and draw the L.P. compression-hyperbola from the point 14, located by  $K_2$ . Now turn back to Fig. 41, and note the fundamental fact that if the measure  $pv$  of the working-steam is to be the same in both cylinders, distances between hyperbolas of expansion and of compression, along any pressure-line, must be the same for both diagrams. It is pretty evident that the H.P. clearance-steam is greater in quantity than the L.P., or that the curve 9-17 will be outside (to the right) of 16-14: then the expansion-curve 11-12 must be inside of 4-18. Knowing that L.P. cut-off takes place at the ordinate  $C_2$ , we now seek to determine the pressure at this cut-off, or to locate the line 7-11 so that the volumes 17-18 and 16-11 will be equal.

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To do this, we make a rough guess at the position of the point 11, and by trial locate 11 on  $C_2$  so that the area is balanced 16-17. On the return-stroke the cut-off takes place at  $C_2'$ , as marked. The gas is then compressed in the H.P. cylinder and the cycle is closed at  $K_1$ . For the curve of this compression we assume  $O'$ . By a single construction we get the position of 18, and then by another construction with the same expansion-ratio we get the compression-curve down to 17. If this does not agree with 11-18, we try again, until the two curves are equal.

**THE MULTIPLE-EXPANSION CYCLE**

To do this, we make a rough guess at the pressure  $p_1$ , and by trial locate 11 on  $C_1$  corresponding to the pressure assumed 16-17. On the return-stroke cut-off takes place at  $C_1'$ , as marked, and the steam is compressed in the H.P. cylinder and cut off at  $K_1$ . For the curve of this compression we assume  $O'$ . By a single construction we get the pressure  $p_2$ , and then by another construction with  $p_2$  as initial pressure, a compression-curve down to 17. If this does not agree with 11-18, we try again, until the two curves are equal.

Having now the pressure of the steam when it is definitely shut off by itself, we can mix it with the steam released from the L.P. cylinder. One way is to take the measuring volume  $R$  by pressure  $p_2$ , and change it to  $p_1$ . Just after the valve opens at 4, the steam fills the amount  $P'-4$ ; but a piston would fail to fill this space by the time the pressure had reached this deficiency to 25-4, and drawing the tangent at this point as pole, we get two results: first, that the steam expands tangentially and widely, the resultant pressure being  $p_{26}$ ; second, that this mixture would be admitted at the instant of L.P. admission at ordinate 25 in the H.P. cylinder will follow some curve which can easily be sketched in by eye.



and divide it into, say, 10 equal parts, drawing ordinates as indicated; and on this ( $V_2 - V_1$ ) base locate the L.P. cut-off ordinate  $C_2''$ . The point G thus found will correspond with 11 or 7, and the curve GF has O as its pole. Now divide  $M_2N_2$  and  $N_1M_1$  into the same ten equal parts, and project over heights found for FG to the corresponding ordinates, to get 6-7 and 10-11.

The rest of the operation is self-evident. Curves 9-1, 5-6, 15-10, and 12-13 are sketched in so as to look most natural. Measurement shows that the L.P. diagram-area  $A_2$  is about 1.24 times as large as the H.P. area  $A_1$ , or 134679; but the further secondary effects not here taken into account—namely, friction of the steam-passages and collapse of the steam in passing to the L.P. cylinder—all tend to reduce the relative value of  $A_2$ , and this will probably cause the actual work to be almost equally divided between the pistons.

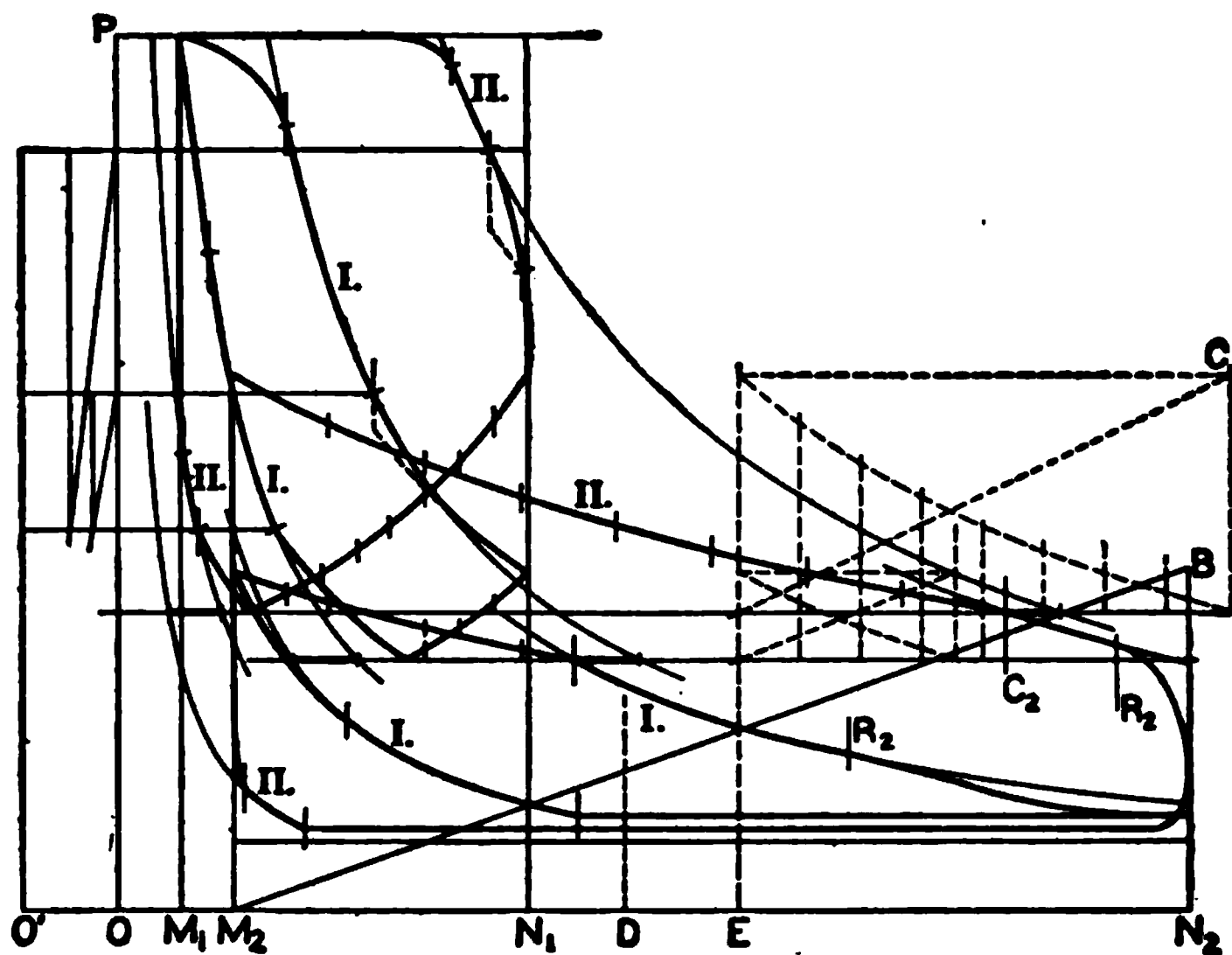


FIG. 581.—Earlier and Later Cut-off, in Engine of Fig. 580.

(f) THE EFFECT OF VARIABLE VALVE-ACTION in this locomotive is illustrated in Fig. 581, where diagrams are drawn for H.P. effective cut-off at one-quarter and at three-quarters of the stroke,

with the steam-distribution determined by Fig. 582. An important point in the design of any variable cut-off compound engine which has both exhaust and admission controlled by the same valve, is to keep the compression from running too high, especially in the H.P. cylinder. Here this end is attained through the use of large clearances and negative exhaust-laps. It appears from Fig. 580 that the resulting early release has no bad effect on the H.P. diagram, but that the exhaust begins much sooner in the L.P. cylinder than is desirable. To show that more lap can be used to advantage, Figs. 581 and 582 are drawn with an exhaust-lap equal to zero, instead of minus  $\frac{1}{2}$ " on the L.P. valve-face. Even with the earliest cut-off here investigated, it does not appear that either compression is too great. The exhaust-pressure is made higher for the smaller diagram because the speed is supposed to be very much greater.

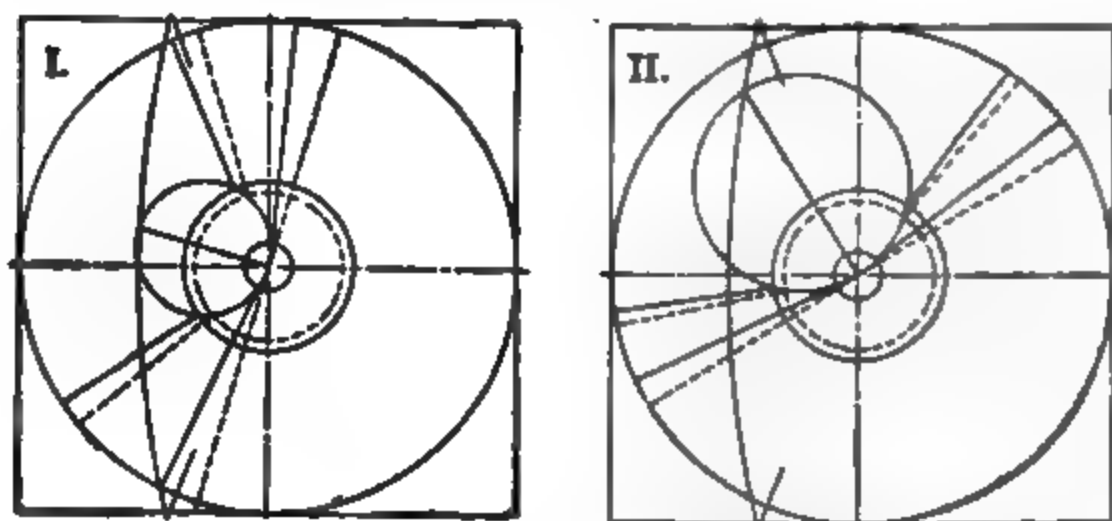


FIG. 582.—Valve-diagrams with Fig. 581;  
full lines, H.P. cylinder; dotted lines, L.P. cylinder.

The construction of Fig. 581 is the same throughout as that of Fig. 580. Note that in diagram I, the ideal curve of expansion after the mixing at H.P. release rises above the main expansion-hyperbola. The ratio of  $A_2$  to  $A_1$  in this figure is 0.96 to 1.00 for Case I., 1.7 to 1.0 for Case II.

(g) INDICATOR DIAGRAMS, original and combined, from a locomotive like that for which Figs. 580 and 581 were drawn, are shown in Fig. 583, with the idea of comparing actual with pre-

dicted results. These diagrams are the mean of cards from both ends of the cylinders on one side of the locomotive. The valve-action is practically the same as in Fig. 580 except that there is more exhaust-lap, and the various "events" and curves of the two

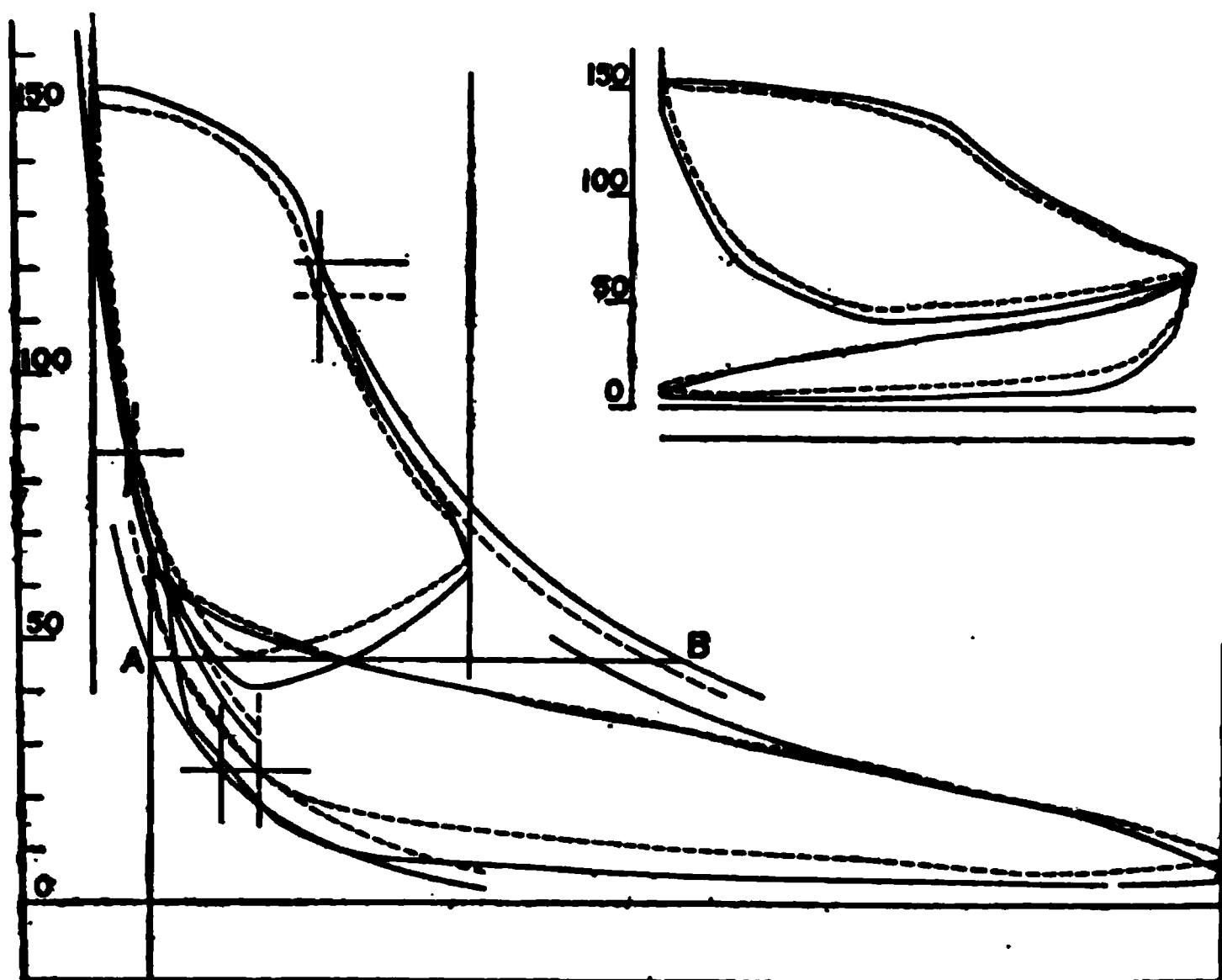


FIG. 583.—Indicator Diagrams from Vaucain Locomotive.

diagrams are essentially the same in timing and character: one point to be noted is the prompt flow from H.P. to L.P. when the latter opens for admission, causing a rapid admission-rise and a small loop on the L.P. diagram.

The greatest differences are due to the throttling action of the ports and valves, by no means sufficiently allowed for in Fig. 580. The two sets of diagrams in Fig. 583 were taken at different speeds, the full lines at 40 miles per hour, the dotted lines at 60 miles. A matter of great interest is the variation in the back-pressure in both cylinders, with practically the same curves of forward-pressure at both speeds. With the higher speed, the exhaust-pressure has a minimum at about 6 lbs. above atmosphere and rises to about

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where compression begins—th  
definitely marked.

chiefly on account of this modifi  
tting action, the work-division i  
Fig. 580, being here almost exactly  
steam-quantities, measurements along  
bolas, give the ratio 0.92 for the full  
dotted-line, this being the ratio of th  
t this pressure; that is, the "collapse  
8 per cent. and 5 per cent. in the re  
Some further examples of diagrams  
expansion type will be found in § 6  
mechanical action but also the questi  
are taken up. We now pass, however,  
cated case of the engine with non-simul  
most emphasis on the study of the resu  
ideal pressure-volume analysis.

### § 68. The Receiver-com

(a) THE QUARTER-CRANK COMPOUN  
in § 22 (c), the term "receiver-compou

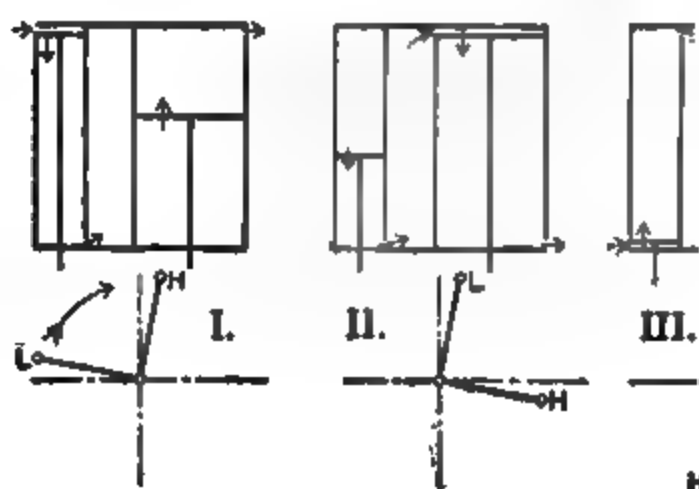


FIG. 584.—Action of the Quarter

strictly distinctive, is about as short an  
designed for the class of engines in which  
and end together, so that a receiver betw

lutely essential to successful working. Of this class, the engine with cranks at right-angles is the most common and important type.

The action of the steam in this engine is illustrated in general terms by the series of position-diagrams in Fig. 584. The space between the cylinders represents the receiver, and it is assumed that the cut-off takes place not later than at half-stroke in either cylinder. Diagram I. makes it evident that the steam released from the lower end of the small cylinder will have to be compressed, in cylinder and receiver, by the advancing H.P. piston; this condition lasts till mid-stroke, when the L.P. piston gets to its top end and the large cylinder starts to draw from the receiver. Position II. shows common expansion in the two cylinders and the receiver, this continuing to L.P. cut-off. The last two diagrams exhibit similar conditions for the other pair of successive cylinder-ends. Note that with the H.P. crank leading, as here, steam from one end of the H.P.

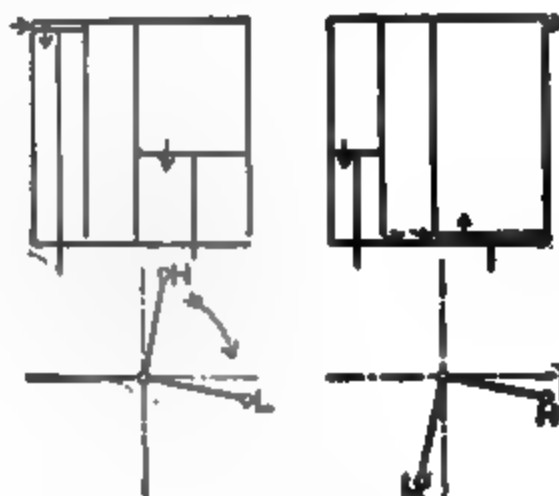


FIG. 585.—Engine with Low-pressure Crank Leading.

cylinder passes into the opposite end of the L.P. Fig. 585 shows, however, that when the L.P. crank leads, the same ends (both "head" or both "crank") are in series. This determines the manner in which actual indicator diagrams are to be combined; but it must be remembered that, in any engine with a common receiver, the division of steam between the ends is not necessarily the same in successive cylinders; and to get entirely definite results an average combination is required, as exemplified in Fig. 41 and by the diagrams in § 69.

(b) VARIATION IN THE STEAM-VOLUME.—This can be most fully shown and clearly followed by the method of Fig. 586. The vertical base of this diagram is the developed crank-circle, representing also the time of one revolution; the horizontal ordinates are volumes. The primary volumes  $V_1$ ,  $R$ , and  $V_2$  are laid off as

indicated; and by plotting sinusoids in proper relation (see Fig. 12) measure of the varying volume. Here and the connecting-rods are taken as

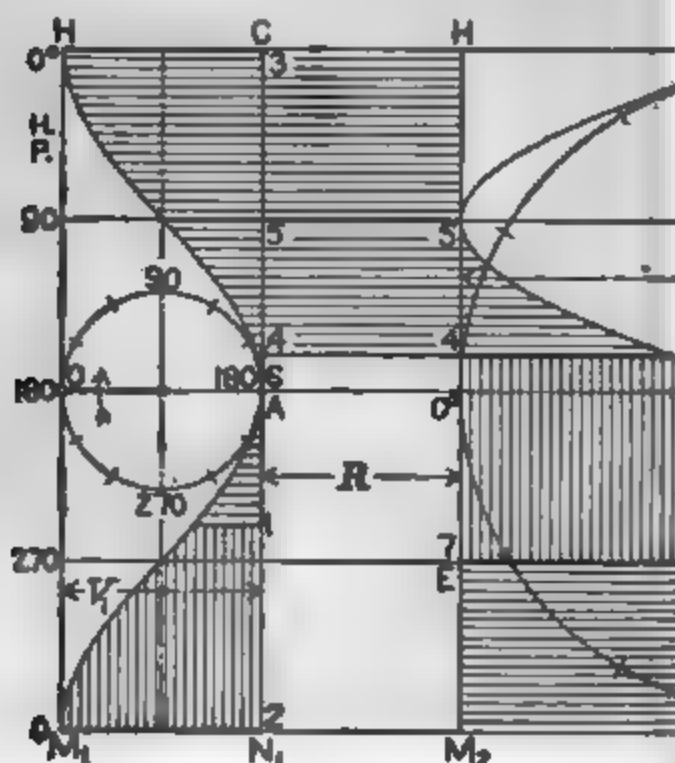


FIG. 586.—The Volume

The action represented by the sh designated by reference-numbers identi and 589, takes place in the H.P. crank-since, for illustrative purposes, this coi the way the curves are drawn. It beg A, lasting to the cut-off at 1; then foll of stroke at 2. Here the valve opens t den increase in the volume involved (a bodies of steam. From 3 to 5 this cor in H.P. cylinder and receiver, until L Even after 5 there is at first a decrea small piston is moving so much the m

\* Horizontal shading shows an action dur main volumes are in communication; vertica the body of steam is definitely confined in one

expansion really begins, and continues to L.P. cut-off at 4. Thereafter, the working-steam expands in the large cylinder to 7, then exhausts; while the receiver-steam is slightly compressed by the H.P. piston, from 4 to 6.

(c) **PRESSURE-VOLUME DIAGRAMS** for this type of engine, analogous to Fig. 579, are drawn in Fig. 589, for the two distinctive cases as to the intermediate action between the cylinders. The full-line curves correspond with the conditions in the preceding figure, where the cut-offs are early; the dotted-line diagrams show the effect of late cut-off. The important distinction lies in the question whether L.P. cut-off occurs before or after half-stroke

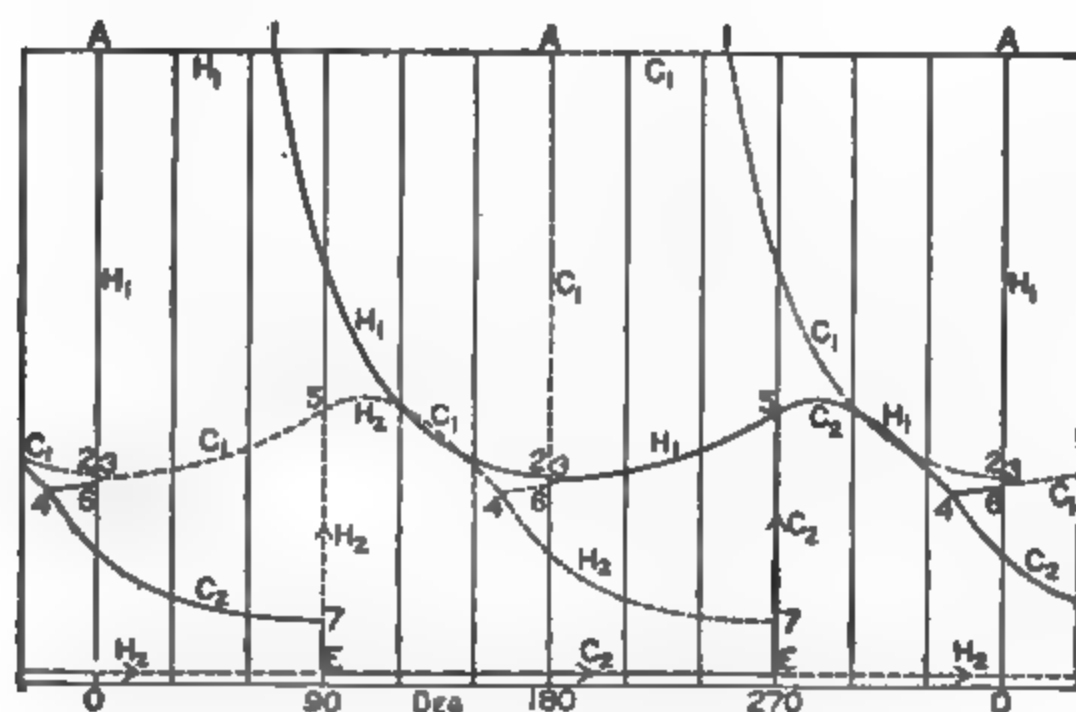


FIG. 587.—Pressures on a Time-base, Case A.

—that being taken as the point where the H.P. cylinder releases. As to general proportions, Fig. 589 has the volumes  $V_1$  and  $V_2$  in the ratio that would be given by the diameters 20 and 36, or as 1 to 3.24; and the receiver-volume  $R$  is made equal to the H.P. cylinder. The cut-offs, the same in both cylinders, are at one-third and at two-thirds of the stroke in cases A and B respectively. Similar points on the two diagrams have the same numbers, some of these being, however, further distinguished by the letters A and B.

It will be much easier to follow Fig. 589 after first running over Fig. 587 and 588. These are plotted on a circle for a base—the degree-scale below with zero at the head-end dead-cent

The curves simply show how the form closed figures, nor are the areas work. This system of representation L.P. diagrams can be fitted under pressures on the same ordinate.

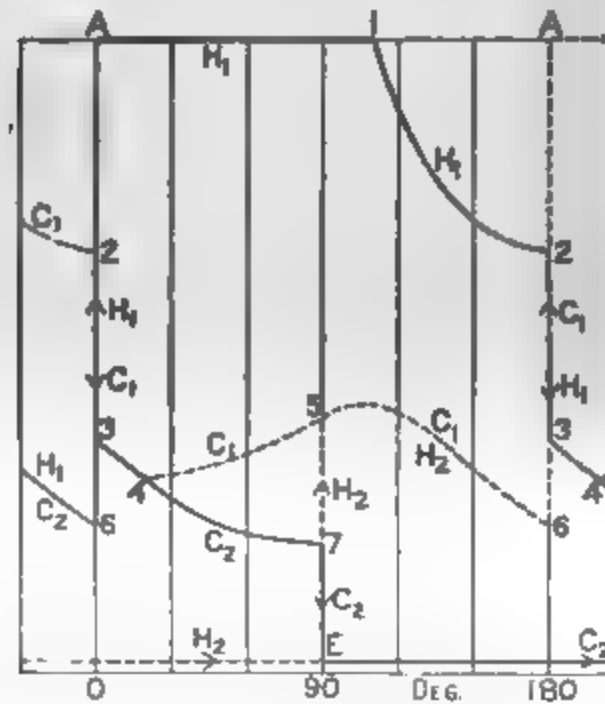


FIG. 588.—Pressures on a 7

In Fig. 587 the sequence of events admission, A to 1; expansion, 1 to 2; receiver-steam which was at 6, giving at 3; compression in H P. cylinder at 3; compression in H P. cylinder at 3; admission at 5; common expansion, 4 to 7, with exhaust at E; and receiver, 4 to 6. On the figures, 1 C for crank-end, with subscripts 1 and

In Fig. 588 the difference is that wh



2, the receiver is yet open to the L.P. cylinder: there is there-  
re a short period of common expansion, to L.P. cut-off at 4;  
hereafter, compression into the receiver raises the pressure to 5,  
here admission to the other end of the L.P. cylinder begins.  
his lasts to the end of the H.P. stroke, dropping the pressure to  
whence it is suddenly raised to 3 by a fresh exhaust from the  
ther end of the H.P. cylinder.

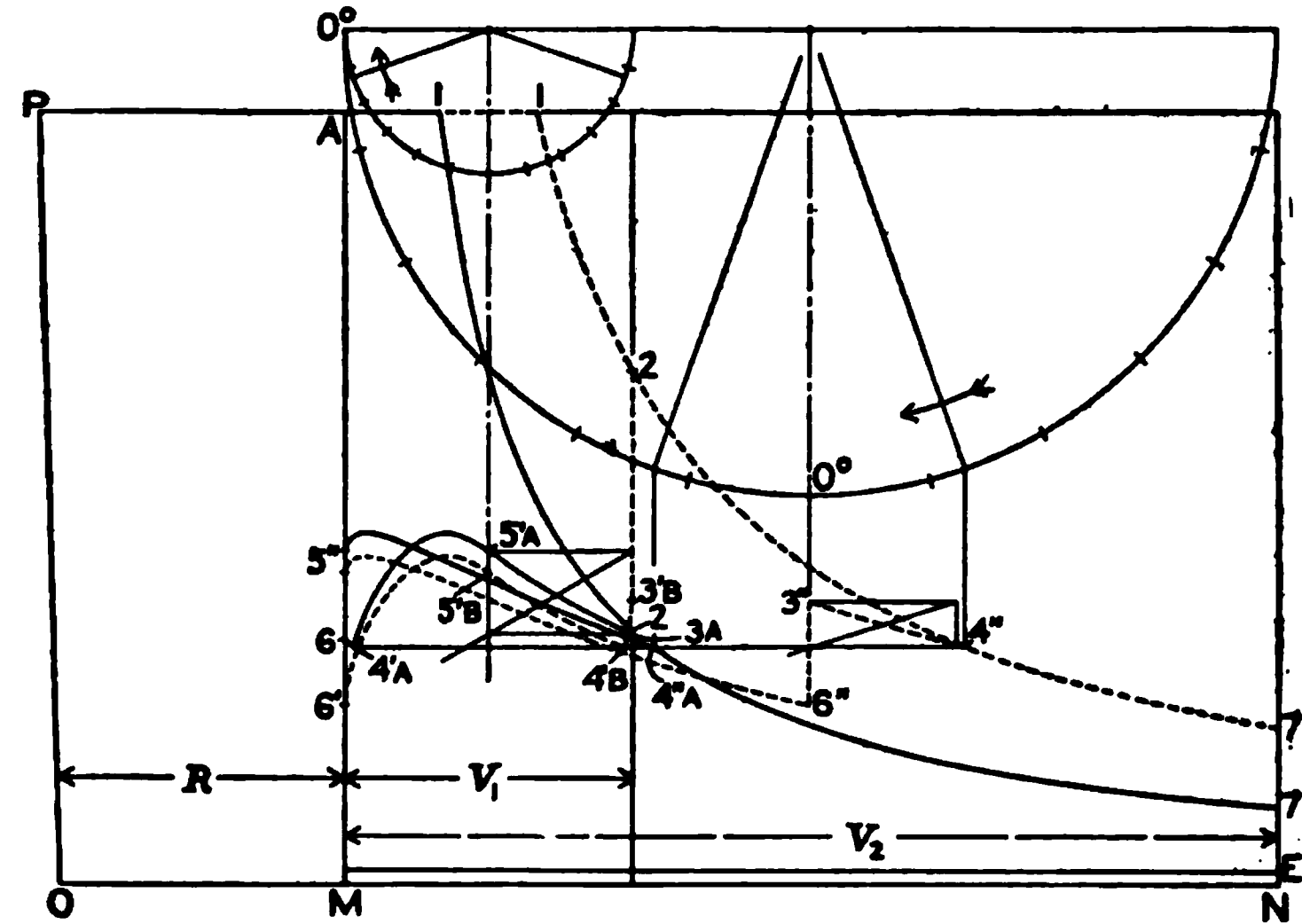


FIG. 589.—Pressure-volume Diagrams.

(d) DRAWING THE PRESSURE-VOLUME DIAGRAM.—Coming now  
to the construction of Fig. 589, we as usual have the L.P. cut-off,  
on the main expansion-hyperbola, for a starting-point in the  
derivation of the intermediate curves. In Case A, with early  
cut-offs as in Fig. 587, curve 45 is first laid out from 4 backward  
to 5; the method is more closely shown by Fig. 590, which is a  
part of Fig. 589 enlarged. Here C<sub>2</sub> is the L.P. cut-off 4'' and C<sub>1</sub>  
is the corresponding position of the H.P. piston, or the point 4';  
then at the pressure p<sub>4</sub> the total volume v<sub>4</sub> has the two parts indi-  
cated on the figure; and these, added together, make v<sub>4</sub> extend to  
the ordinate through A, which is one of the secondary axes for  
the hyperbola-construction. For any other pair of simultaneous

iston-positions, as  $P_1$  and  $P_2$ , additional lines in H.P. cylinder and receiver

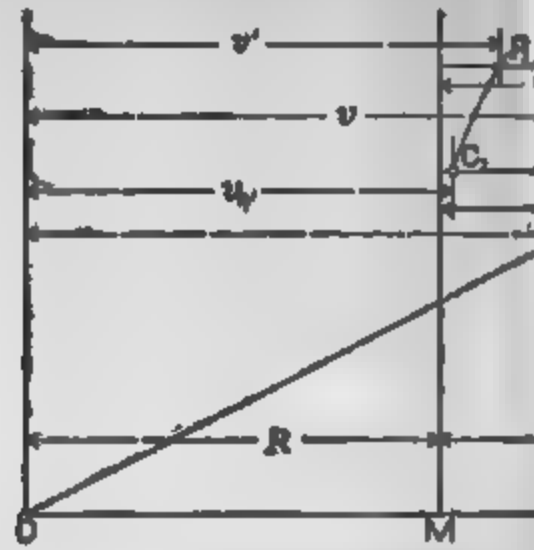


FIG. 590.—Intermediate

gives the total volume  $v$  and the po from O determines the pressure by jected over, this locates vertically th on the respective 45 curves. Having 5, on Fig. 589, we next get 5'3 as pole, thus finding the pressure at 3 gives the short curve 4'6.

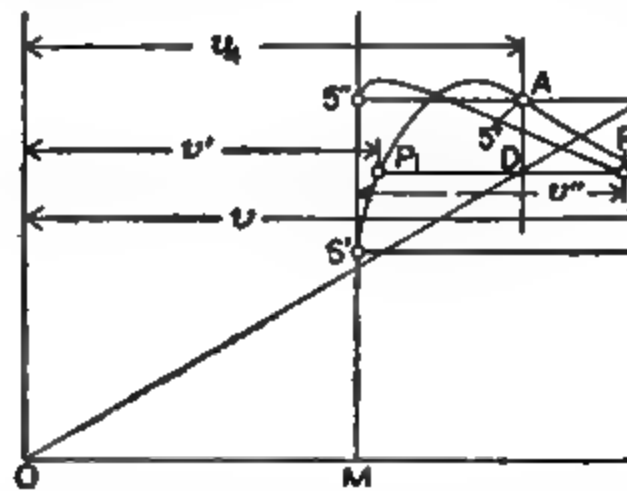


FIG. 591.—Intermediate

For Case B the operation is a lit 591, with L P. cut-off at  $C_2$ , the H.P.

way on the return stroke, and from  $C_1$  the curve of compression is carried up to A or  $5'$  to get a starting-point for the line of common expansion. Having AB and AD as axes, we add volumes, find pressure as at D, and project over to the position-ordinates to determine  $P_1$  and  $P_2$ . The same method is used for the short curve C3 or 43.

The greatest difference in steam-effect between this and the direct-expansion type of engine is found in the pressure-curve for the H.P. return stroke, which rises at the middle in one case, falls in the other. Further illustration is given in Fig. 592, where cut-off is varied in the L.P. cylinder only. The simplest case is seen where the latter cut-off is just at mid-stroke, for then the H.P. back-pressure curve has only two parts or phases, as against

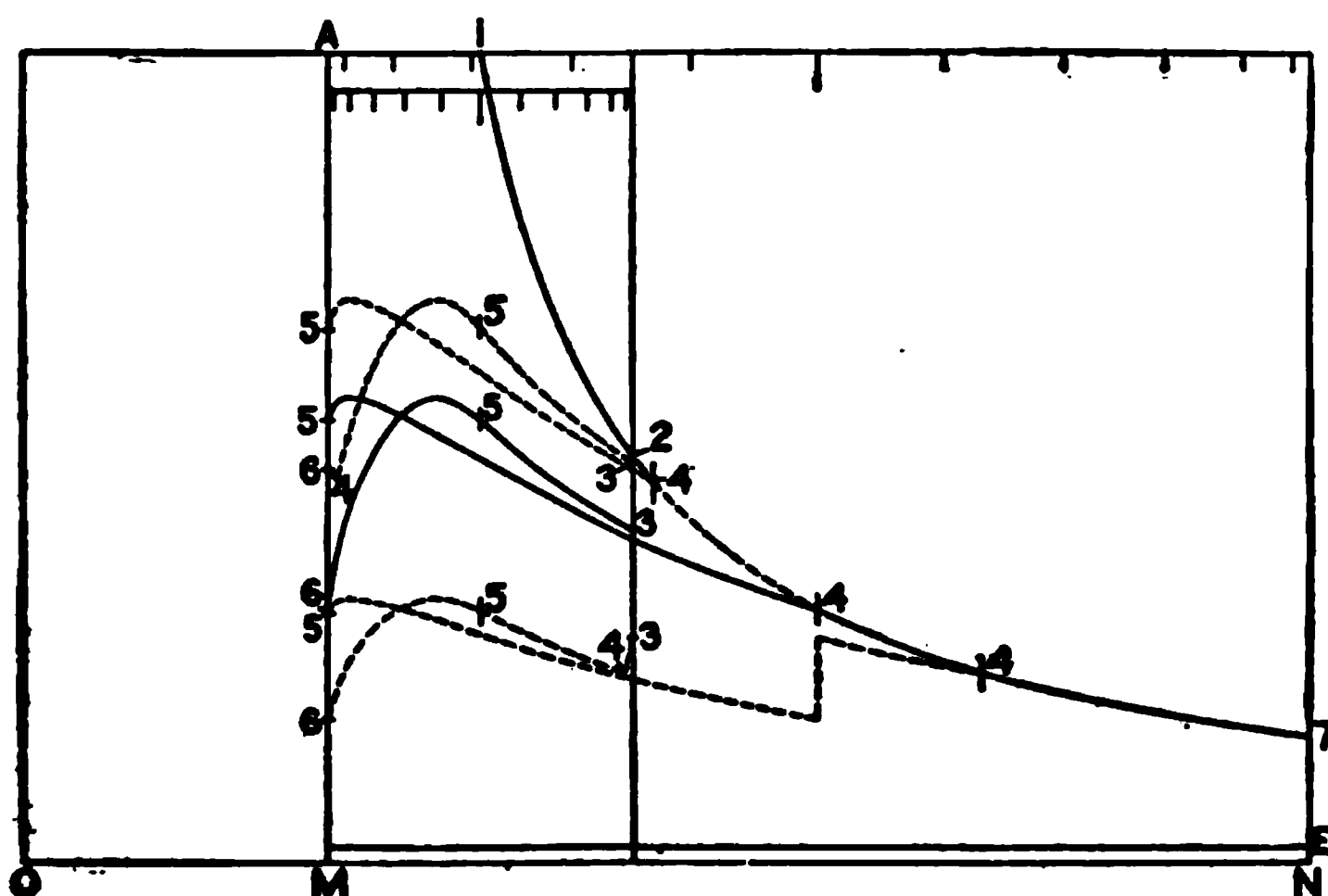


FIG. 592.—Variation in L.P. Cut-off.

three in the other cases. The general effect upon the receiver-pressure, due to varying the L.P. cut-off, is the same as was described in § 66 (c); but with the rather small receiver here used, the pressure-changes throughout the stroke are very great. To get a better idea of what to expect in an actual engine, we will now analyze an example with a larger receiver, taking into account the clearances and, approximately, the valve-action, as in Fig. 580.

(e) ANALYSIS OF AN ACTUAL ENGINE. The proportions suitable to use are those of the engine, or, if they are not known, they may be assumed, chiefly because this will make all the curves of the diagram large enough to be visible even on a reduced scale. The proportions are, in fact, practically identical with those of the receiver; as used in laying out the diagram, the receiver is taken in terms of the L.P. cylinder as unity, the

$V_1$	$C_1$	$R$
.36	.07	.54

the receiver being one and one-half cylinders. The valve-action, with the receiver, is shown in Figs. 580 and 582, but with both ex-

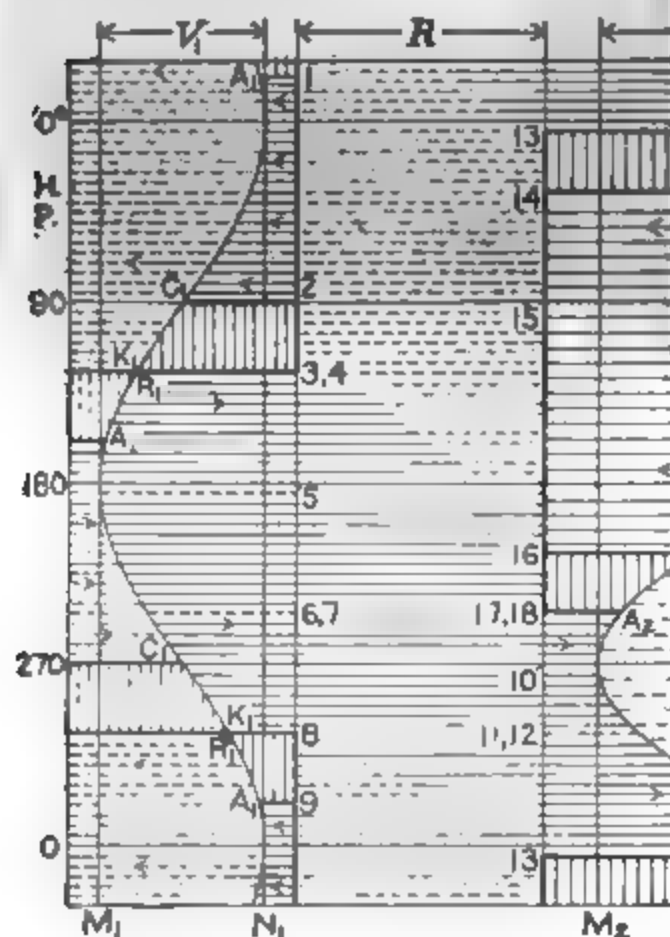


FIG. 593.—Volume Diagram 1

grammed at Fig. 594 II., and the e properly located on the volume-diagram diagram, Fig. 594 I., with the same de

In Fig. 593, full-line shading is used for the "cycle" made up the operations in H.P. crank-end and L.P. head-end, dotted shading for the other cycle. Merging of these two cycles, where occurs, is indicated by running together the two kinds of shading the receiver-space. Beginning at point 1, we have H.P. admission to cut-off at 2, then expansion to 3 (follow the operation on Fig. 594 also). When release takes place at 3, the receiver is open to the L.P. end which belongs to the other cycle—the same effect being there produced that this cycle receives at 11, 12. As the result of the mixing, the H.P. steam drops from 3 to 4, the L.P. steam rises from 11 to 12. On Fig. 594, all these intermediate changes are represented, by the full black lines, as if they resulted from a full, immediate valve-action, with instantaneous equalization of pressure throughout the spaces in communication: and the dotted curves show a guess at what actually would take place.

After equalization at 4, there is common expansion till L.P. cut-off at 5, then compression in H.P. cylinder and receiver to L.P. admission at 6. Here, with H.P. drop from 6 to 7 and L.P. rise from 17 to 18, really begins the L.P. portion of this cycle. From nearly to 8 (or from 18 past 10), there is compression; at 8 the H.P. exhaust closes, and at the same time the other end of the H.P. cylinder releases, causing the rise from 11 to 12, as already explained. It will be noted that on account of the very early release from the H.P. cylinder, this engine is decidedly in the class typified by Fig. 588, even though the L.P. cut-off occurs very soon after half-stroke.

(f) CONSTRUCTION OF FIG. 594.—With Fig. 593 to make clear the course of events, it is not difficult to trace out fully the whole intermediate action between the cylinders; but we have here a case where the purely graphical method must, for ease and clearness, yield to a method of measurement and computation.

The L.P. cut-off is, of course, the starting-point. Just as in Fig. 580, we can see that the H.P. compression-hyperbola will be outside of the L.P. curve 22-16, hence 13-14 must be below or inside of 3-19. The pressure at 13 which will make 22-13 equal 21-19 is computed as follows.

At point 20,  $p=62.5$ ,  $v=0.970$ ,  $v_1=1.635$ ; these volumes being in

For cylinder-steam alone (we gives  $pv=62.5 \times 0.665=41.56$  for clearance-steam, taking co-ordinates  $pv=18 \times 0.345=6.22$ .

Now let  $p$  be the unknown pressure, volumes at 19 and 22 we have the  $6.22/p$ .

After the L.P. cut-off, steam in the cylinder along 56, to the next L.P. admission is compressed to 0.866. Here this steam,  $p \times 0.970$  in L.P. clearance-steam, and the result  $(p \times 0.970 + 6.22)$  with the pressure further compressed to H.P. exhaust-pressure is 0.148 in H. plus 0.742 in R. and the pressure at this point 8 is

$$\frac{p \times 0.970 + 6.22}{0.890}$$

multiplying this by the volume 0.148 in H.P. clearance steam; and division by the volume at 21.

The four volumes along the line expressed in terms of  $p$ , we have

$$v_{12} - v_{22} = v_{13} - v_{23} \\ 0.665 - \frac{6.22}{p} = \frac{41.56}{p} - \frac{0.148}{p}$$

This gives

$$0.665p - 6.22 = 41.56 - 0.148$$

$$p = 56.64.$$

Knowing this  $p$ , all the critical pressures are as indicated above; and the same method of getting a series of points on the several expansion curves. For example, on expansion 4-5, 12-13, we have the pressure at 13,  $p=56.64$ , at 13. Measuring volume

the results in Table 68 A. Similar calculations are used for curves 7-8, 18-10-11. The only plain hyperbola, among all these intermediate curves, is 56, with a pole located by measuring off the receiver-volume  $R$  to the left from  $O$ . Of course, the constructions shown in Figs. 590 and 591 might be used here; but the other

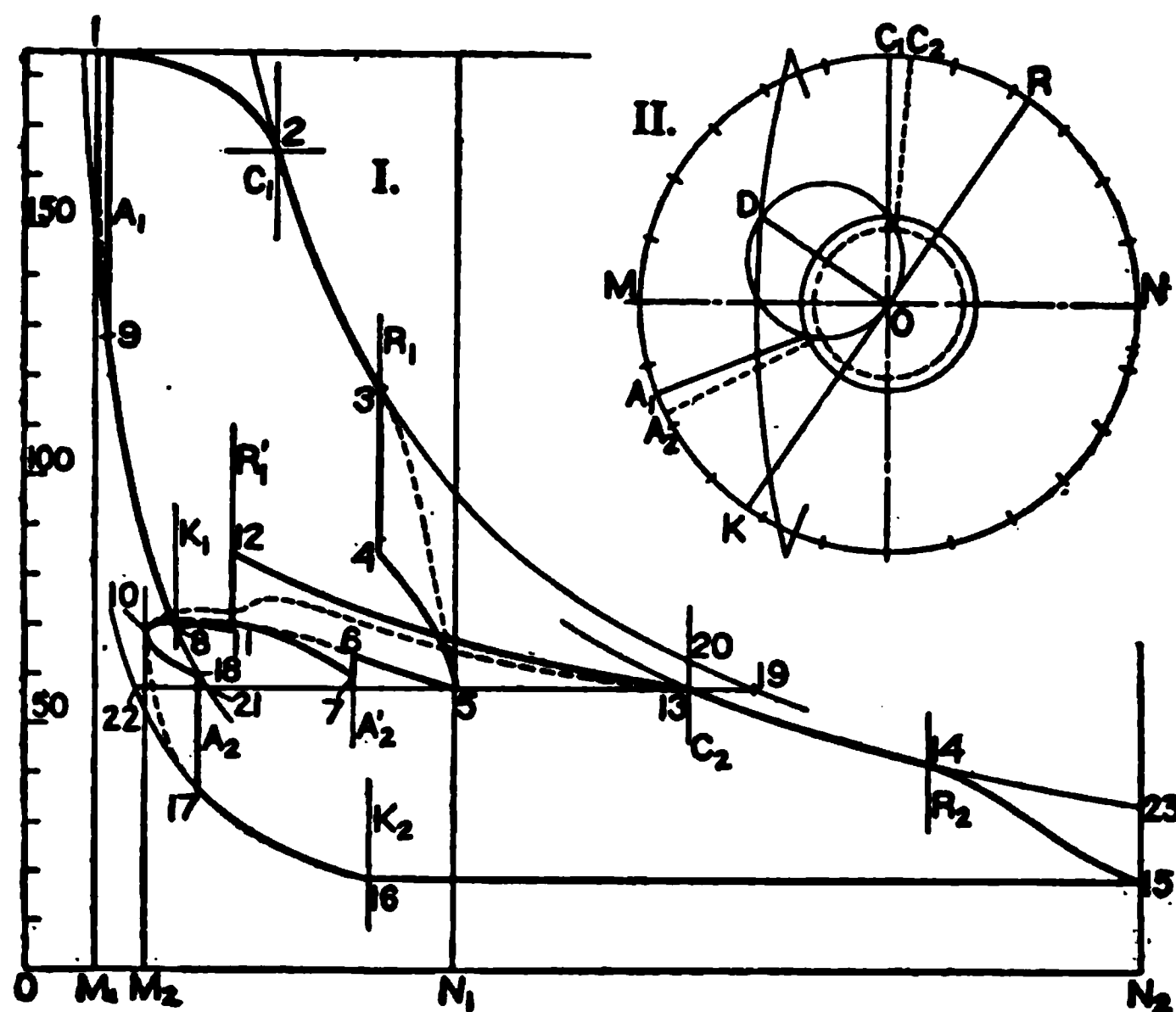


FIG. 594.—Ideal Steam-diagram for Real Engine.

method seems preferable, especially with the slide-rule for computation. To get  $p$  at 13, the determination by calculation is overwhelmingly preferable to graphical determination by successive trials.

TABLE 68 A. POINTS ON CURVES 4-5, 12-13.

	$H+R$	$L$	
At H.P. release	$v=0.892+0.208=1.100$		$p=84.2$
" 135° H.P.	$v=0.917+0.267=1.184$		$p=78.2$
" 150° "	$v=0.946+0.370=1.316$		$p=70.4$
" 165° "	$v=0.964+0.491=1.455$		$p=63.2$
" 180° "	$v=0.970+0.620=1.590$		$p=58.3$

## THE MULTIPLE-EXPA

(g) GENERAL CHARACTER OF THE P  
arked in (d), the distinguishing a  
compound-engine diagram is the .  
the H.P. cylinder, which drops toward  
ous strokes, but rises at the middle  
angles. We shall fully enough cover  
by considering further what happens

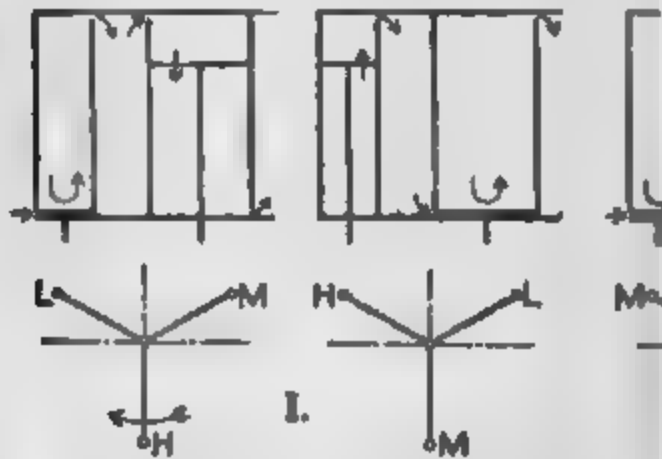


FIG. 595.—Different Arrangements of

In Fig. 595 are illustrated the relative stages of a triple-expansion engine, only cylinders being represented in the sketches. In the first arrangement, the first with the order H M L, the other with low or L M H or H-L M. In the first case, H has completed its stroke and exhaust is at the end of the stroke; piston M is at about quarter-stroke. There will be, therefore, first a gradual drop, then a rise as piston H continues its stroke; when piston M gets to the beginning of its stroke and begins to draw steam from the receiver, the drop will show on the H diagram depression begins in that cylinder. A few examples of this arrangement are given at 607. Complete suppression of the final drop at II. it appears only as a slight change in the rapid upward slant near mid-stroke; in



being given more time to develop, but is much diminished in amount because the receiver is very large.

Of the other arrangement, outlined in Fig. 595 II., only one actual example is given, in Fig. 605 II. Here the receiver is closed when cylinder H exhausts, piston M being at about three-quarters stroke; but cylinder M soon starts to draw (sketch 2) and remains open till piston H gets well toward the end of its stroke. The retarding effect of the passages will account for the fact that the drop of the H exhaust-line (on Fig. 605 II.) begins somewhat later in the stroke than it would under the ideal conditions which have been assumed in previous discussions.

(h) VALUE OF THE VOLUMETRIC ANALYSIS.—Increasing familiarity with actual diagrams will develop and strengthen the conviction that the detailed analyses given in Figs. 580 and 594 are likely to be useful rather as leading to a thorough understanding of the character of the interactions between the cylinders than as practical methods to be closely followed when seeking quantitative results. Throttling effects, evidenced by the rounding of corners and the smoothing out of (ideally) abrupt changes, and by the pressure-gap between the successive stages, together with departures from the law  $pv = C$  due to varying thermal reactions, decidedly modify results got by the methods just referred to—a comparison of Figs. 580 and 583 illustrating these statements very forcibly. The influence of steam-passages, cylinder-walls, jackets and reheaters, etc., cannot be reduced to numerical expression, but can be estimated and predicted only by analogy. The method that suggests itself, then, for the design of the cylinders of a multiple-expansion engine, is about as follows:

Start with an elementary diagram like Fig. 577, and sketch in roughly the shape of the intermediate curves, having due regard to crank-arrangement and relative size of receiver, with allowance for pressure-losses.

Introduce the effect of the probable clearance by a change the reverse of that used in getting Fig. 32 from Fig. 31—that is, swing the diagram, originally drawn against the vertical axis, out beyond the compression-hyperbola; this will give actual cylinder volumes.

Having thus found tentative proportions, use the principal

## THE MULTIPLE-EXPAN-

of the preceding pressure-volume diagrams, with various allowances dictated by the probable indicator diagrams for basis for further adjustments and furnish data for the design of the valve.

Actually, in the development of the evolution has taken place by such many designers have had to take for a border-line of well-established empirical border-line is being continually ground back of it more thoroughly and

As an initial contribution to the expert along this line will have to acquire a number of typical combined diagrams will

### § 69. Diagrams from Multiple

(a) GENERAL METHODS.—Nearly all Figs. 596 to 608, have been redrawn of tests of the various engines—in diagrams, in others from the indicator a little less accurate than if laid out. They are all according to the simple lines to a common axis, or like the 1. Where the clearance-effects are large, in engine diagrams, Figs. 596, 607, and expansion and compression curves of brought to immediate, horizontal reference of the volume-measures of the working relative values of the product  $pv$  are

The pressure-volume measures are values given under the letter  $M$  on near total value of  $pv$ , for any point on the expansion curve, the value for the clearance-steam, as compression-curve. Then the net  $pv$  H.P. cut-off is taken as unity, and the terms of this unit.

Similarly, under the letter D are given sets of numbers showing the work-division among the several cylinders or stages, the total work represented by the diagram being taken as unity.

For the sake of a comparison of thermodynamic results, these diagrams are drawn so as to represent the performance of one pound of steam, all to the same scale of pressure, nearly all to the same scale of volumes. This gives some of them, especially those in Fig. 596, a rather awkward shape for the present discussion of mechanical features. The manner of finding the volume-scale can best be explained by means of an example, using the calculations for Fig. 597. Here the steam-consumption was found to be, for the whole test,  $S_H = 15.51$  lbs. per horse-power-hour. For the diagrams combined, the speed was  $N = 20.00$  R.P.M. and the I.H.P. was 266.9. This gives 4140 lbs. of steam per hour, 690 per minute, 3.45 per revolution, 0.863 per cycle, the engine being duplex, with four sets of cylinder-ends and of diagrams. Of the total steam consumed, 11.1 per cent. was condensed in the jackets, the rest used in the cylinders; so that the weight of cylinder-steam per cycle is 0.766 lb. The mean volume of the L.P. cylinder (nominal), or the piston-displacement for one stroke, is 28.67 cu. ft. Now if 28.67 cu. ft. carries 0.766 lb., the proportional volume for 1 lb. is  $28.67 \div 0.766 = 37.36$  cu. ft. This value having been determined for the L.P. diagram, the other volumes are laid off according to the ratios given under the figure. The scales used for both coordinates are chosen with regard to the space available for the figures: with large expansion or a low terminal pressure, the end of the L.P. diagram has to be broken off and transferred to the clear space above.

With only a few exceptions, the diagrams are drawn for one pound of cylinder-steam, or for more than one pound of total steam consumed in a jacketed engine. On each, the saturation-curve for 1 lb. of dry steam is drawn in full-line; and the proportion of steam shown by the indicator can be estimated, at any pressure, by comparing the width of the diagram (or the horizontal distance between expansion-curve and compression-curve), with the distance between the one-pound curve and the vertical axis. This comparison would be made more evident by transferring the

agrams so that their compressi  
pressure-axis, as in the dotted c  
Fig. 609; but the extra labor an  
a small-scale figures are sufficient n

Having the constant-weight cur  
amount of jacket-steam is shown by  
curve for the whole weight of steam  
the top of the figure. The excep  
included within the full-line curve,  
where the purpose is to compare jack  
in Figs. 607 and 608 where the jac  
determined; and in Fig. 606, where  
special.

In every case, the diagrams here  
of a full set of indicator cards—in Figs  
four sets of ordinates are measured t  
the diagrams are the mean of a nu  
representing the whole of a long test.

Under the title of each figure are  
essential dimensions of the engine, as

In general

$D_1$	$D_2$	$D_3$	$\times$	$S$	28"
$d_1$	$d_2$	$d_3$		$V_2/V_1$	2-4"
$c_1$	$c_2$	$c_3$		$V_2/V_1$	.014
$D$ = diameter of piston,					$S =$
$d$ = diameter of piston-rod,					$c =$

When there is a rod on each side of  
are given. In Fig. 597 the L.P. pisto  
pistons have two rods, in order that  
connecting-rod between them. Whe  
in one stage, as in all three parts of  
are hyphenated, and for the correspon  
is used. In a few cases the ratio of th  
cylinder is stated.

Besides these dimensions, the most  
results as to the working of the engin

pressures, which are well enough shown by the diagrams. The meanings of the various symbols are:

$N$  = revolutions per minute—double-strokes, when there is no rotating shaft;

$V$  = mean speed of piston, in feet per minute;

$p_m$  = mean effective pressure, reduced to the L.P. piston after the usual manner—see § 23 (c), Eq. (111);

$H$  = indicated horse-power;

$S_H$  = steam per horse-power-hour;

$J$  = fraction of total steam used by jackets and reheaters;

$I$  = indicated steam consumption, generally expressed as a fraction of the cylinder-steam, but where marked “total” including the jacket-steam in the basis. Of the subscripts, the first, 1, 2, 3, refers to the cylinder, the second, C or R, to cut-off or release.

$t_s$  = number of degrees of superheat.

Most of these data, together with additional information as to thermodynamic performance, will be found collected in the Tables in Chapter XIII. The numbers here used to designate the test show their location in those Tables. Comment on this side of the subject will be reserved for Chapter XIII.

(b) THE COMPOUND LOCOMOTIVE.—The type of action in this engine is well illustrated by Figs. 583 and 596—high pressure of both admission and exhaust, large clearances, and strong throttling effects being the marked characteristics, with a small total ratio of expansion, even when dropping the steam well toward the exhaust-pressure at the effective release. The French engine was chosen for illustration rather than one of several American four-crank designs tested in the same series, partly because it gave smoother indicator cards, chiefly because it has separate valve-gears for the H.P. and the L.P. cylinders, permitting an independent adjustment of the cut-offs. At II. in Fig. 596 the L.P. cut-offs are quite a little later than the H.P., which lowers the receiver-pressure, but gives a work-division of which the advantage is not apparent, unless it be to diminish the amount of work done by the L.P.

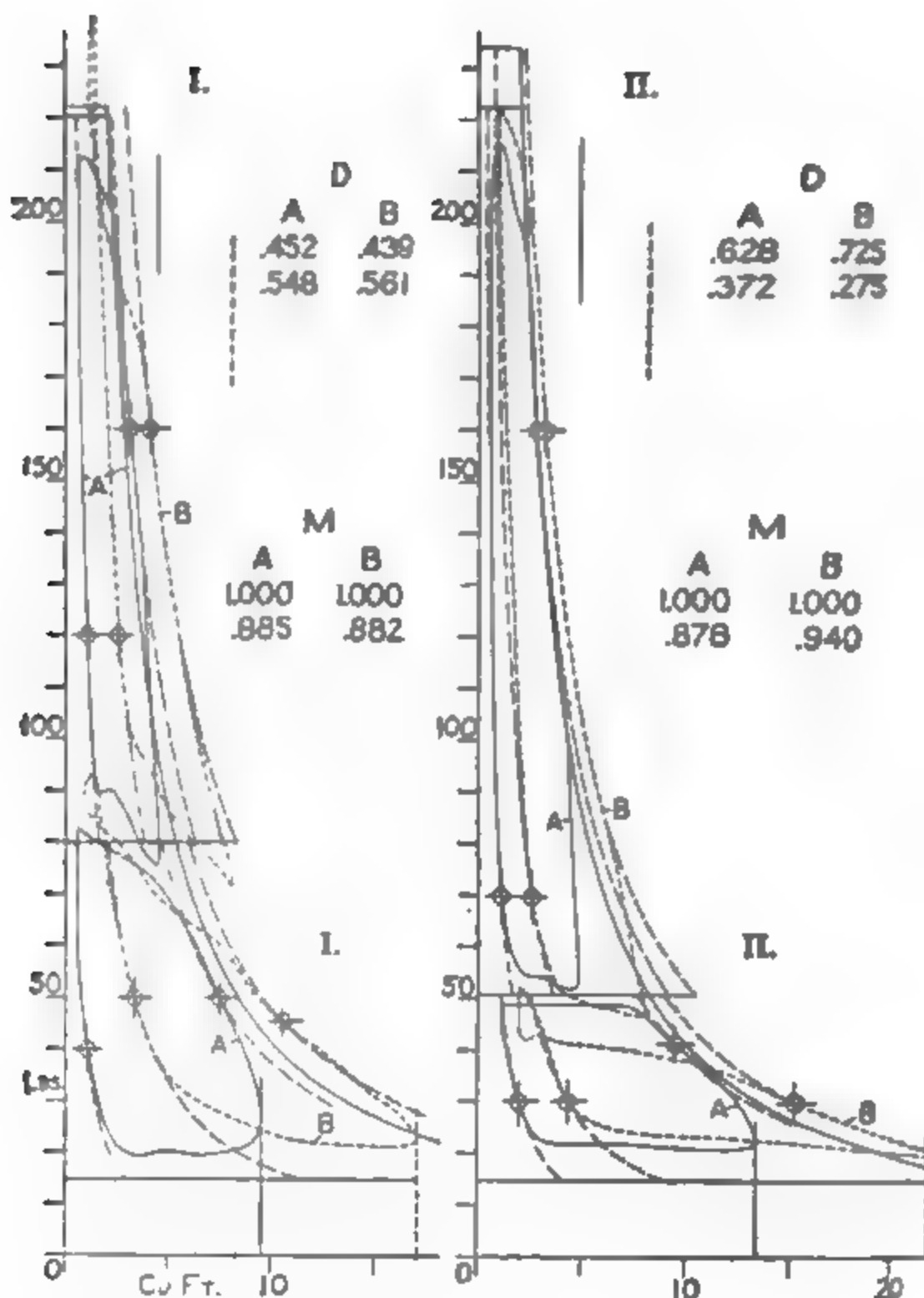


FIG. 596.—Sample Diagrams from Compound Locomotives, tested in the special plant of the Pennsylvania Railroad at the St. Louis Exposition of 1904.

I. Cross-compound (two cylinder)  
Consolidation

23	35	×32
4	4	2.333
.167	.057	$R=2.49V_1$

II. De Glehn Compound (4-cyl.)  
French built, Atlantic type

Two 14 2	23 7	×25 2
2.7	2.7	2.808
.098	.098	$R=1.61V_1$

cylinders, which are between the frames and act upon crank-pins of limited area.

The peculiar shape of the H.P. exhaust-line in II., for an engine with cranks opposite, is due to the fact that the receiver is common to the two sides of the locomotive, the rise toward mid-stroke being caused by inflow from the other H.P. cylinder. The pairs of opposite cranks on the two sides are set at  $90^\circ$  with each other. In I. the cranks are, of course, at right angles.

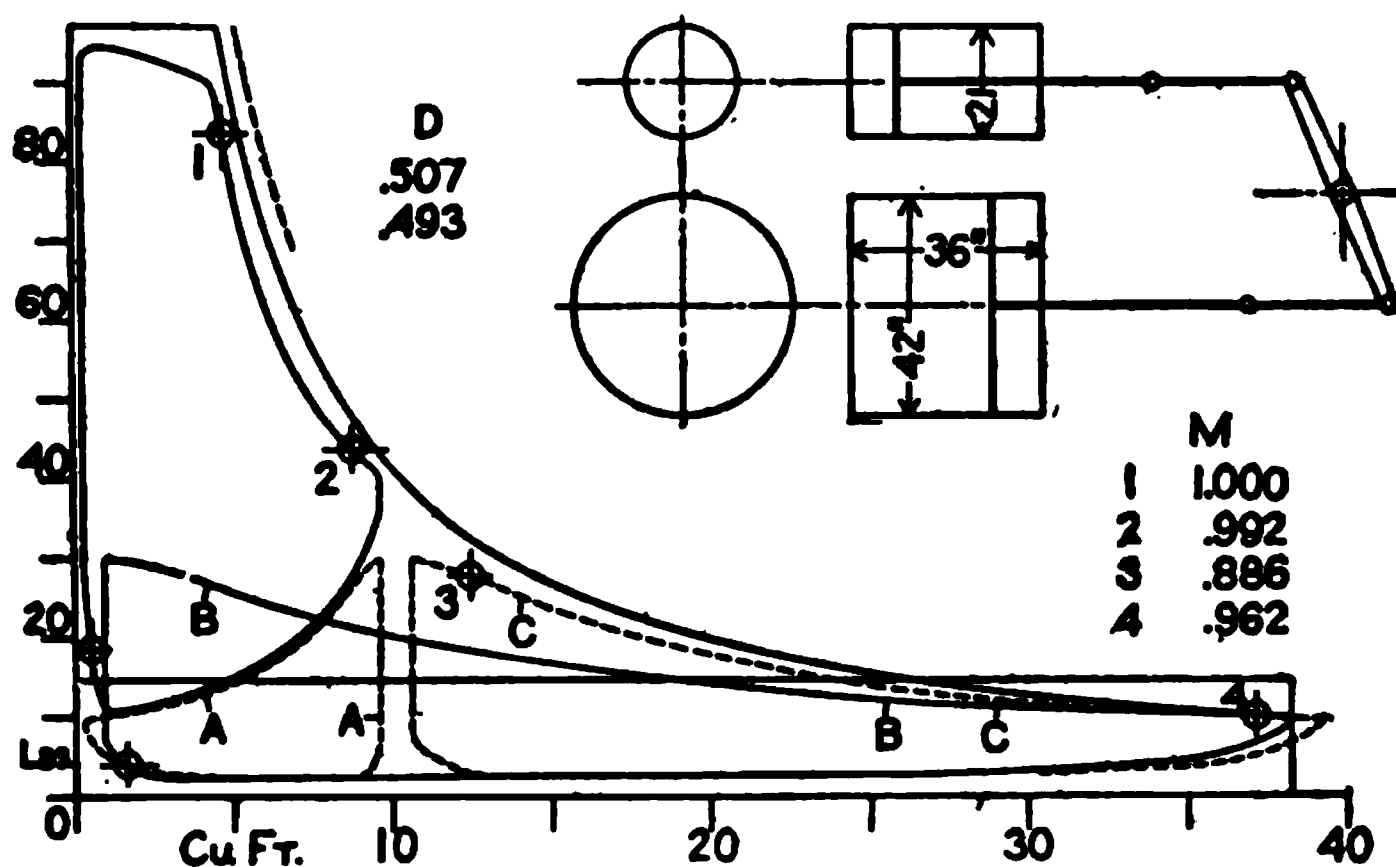


FIG. 597.—Holly (Gaskill) Pumping Engine at South Bethlehem.  
Horizontal, duplex, direct-expansion, no receiver, full jackets.  
May 1902: compare Test 341.

Two 21	42	×36	N=20.00	V=120	$p_m=26.7$
3	2-3½	4.017	H=266.9		$S_H=15.51$
.036	.025		J=0.111	$I_{10}=.870$	$I_H=.955$

(c) LOW-SPEED DIRECT-EXPANSION ENGINES.—Passing now to a radically different class of engines, we take up the four examples in Figs. 597 to 600. The first is an older design of the engine in

FIG. 598—(Continued).

		N	V	$\phi$	$p_m$	H	$S_H$	
I.	A	80.0	427	0.528	75.3	932	19.69	} Compare Test 201
	B	160.0	854	0.496	37.0	890	23.18	
II	A	160.0	672	0.497	52.7	945	20.67	} Compare Test 202:
	B	240.0	1010	0.342	29.8	802	21.62	

Fig. 587 with accuracy in receiver. The low-pressure diagram is drawn in three different ways: first at A, it is placed under the H.P. diagram on the same base-line, closely to show how near the lines of constant expansion agree; next, at B, it is stretched out to the full length representing the volume of the L.P. cylinder as in the last figure; third, at C, and as the best method of showing the equality of the expansion, it is laid off in the form

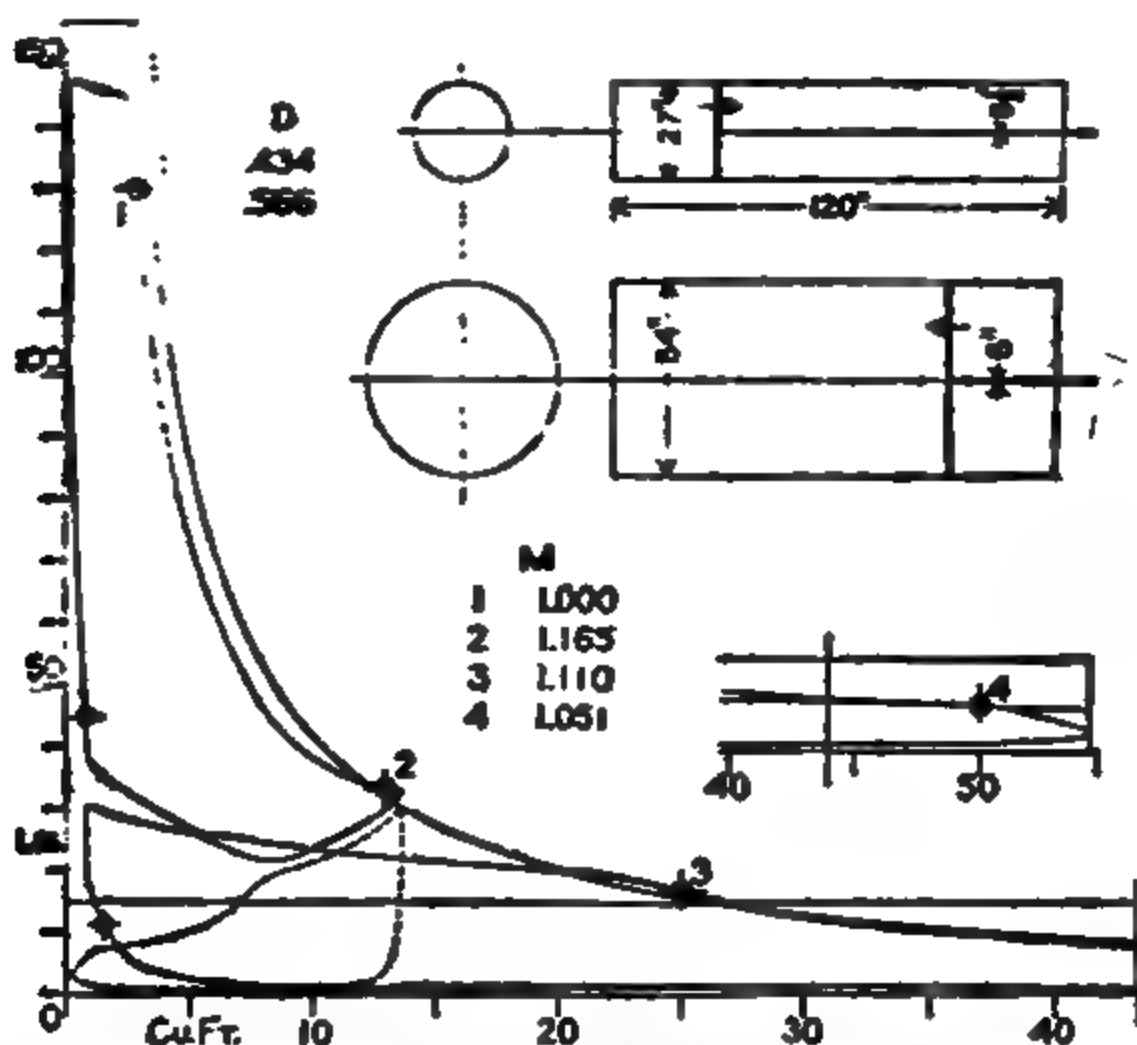


FIG. 598.—Leavitt Pumping Engine at Louisville.

Vertical compound beam-fly-wheel, reheating receiver, full jackets.

April 1894. F. W. Dean, Trans. A.S.M.E., 16-169.

1	54	1.50	V=18.57	V=372	$p_m=24.0$
2	6	1.015	H=643.4		$S_m=12.22$
3	0.153		J=0.167	$I_{ic}=0.766$	$I_m=961$

corresponding to CEFS on Fig. 578. At the beginning of the stroke the connecting rod volume is made both clearances equal axis, and its diagram C starts at this distance f



length is  $(V_2 - V_1)$ . An interpretation of this latter arrangement, from the point of view of work-performance, is that an area  $A_1$  of the L.P. piston receives the work represented by diagram A, and an annular area  $(A_2 - A_1)$  receives the work of diagram C. That is, of course, essentially the same as putting the diagram into the form of E'CEFG on Fig. 578: but here the L.P. clearance must be interposed between A and C, and the presence of a receiver would introduce complications such as would make this method of little utility.

Fig. 598 shows a performance of higher excellence. The combined effect of low speed and full jacketing is seen in the unusually marked re-evaporation toward the end of the H.P. expansion,

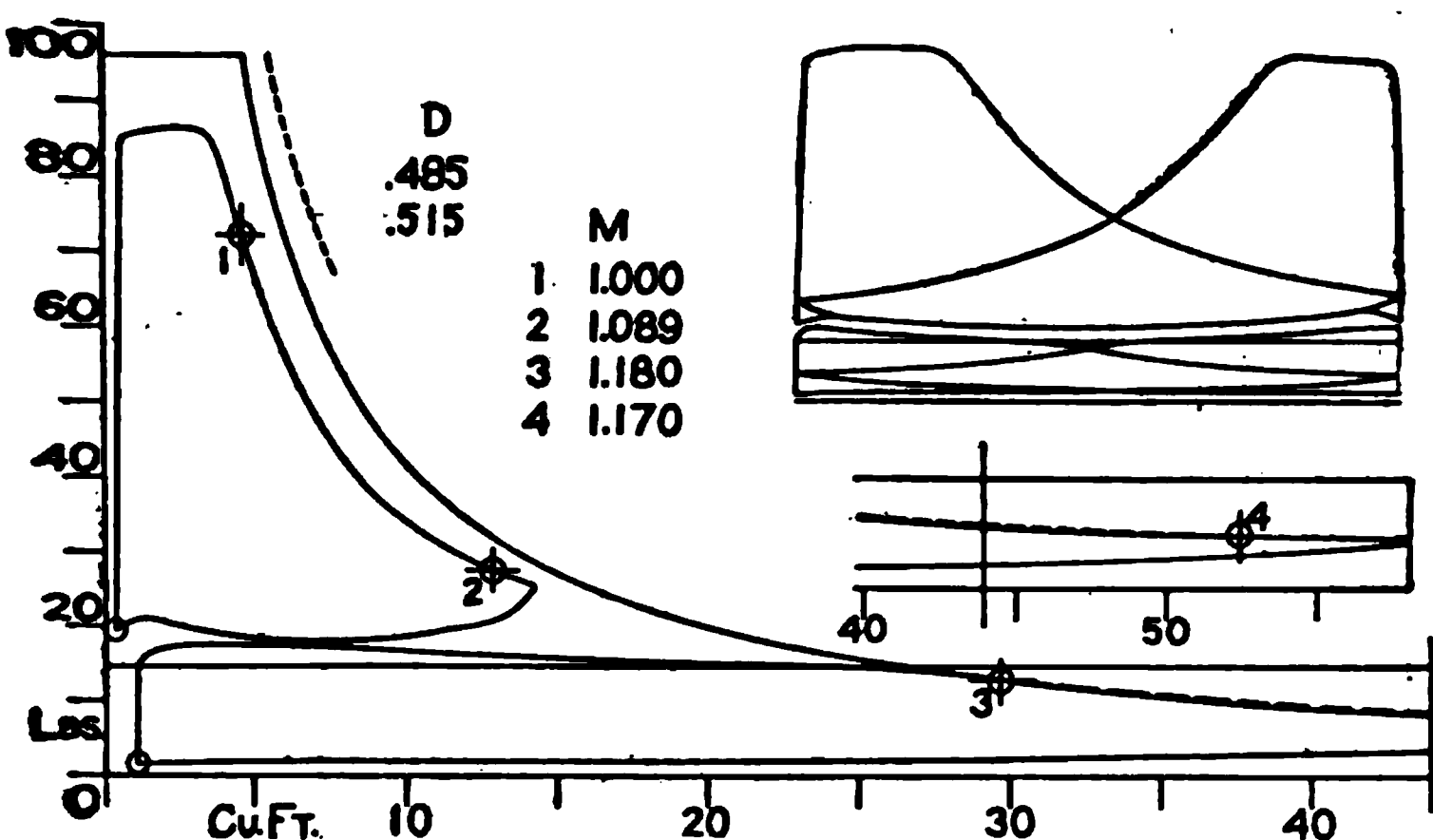


FIG. 599.—Worthington High-duty Pumping Engine, on oil pipe-line.  
Horizontal, duplex, compound, reheating receiver, full jackets.  
Test 337, April, 1891. J. E. Denton, Trans. A.S.M.E., 12-975.

Two 33	66	$\times 37.54$	$N = 20.13$	$V = 126$	$p_m = 17.5$
$5\frac{1}{2}, 5\frac{1}{2}$	$5\frac{1}{2}$	4.10	$H = 456.7$		$S_H = 16.40$
.0251	.0188	$R = 0.7V_1$	$J = 0.161$	$I_{1c} = .788$	$I_{2R} = 1.049$

and in the fact that the apparent or indicated steam is greater in the second than in the first cylinder. The mechanism connecting the two pistons is essentially the same as in the preceding engine, though different in its arrangement and proportions.

Comparing Fig. 599 with 597, we see larger losses by cylinder

condensation in the H.P. cylinder, the diagrams, so that even with a steam-consumption is higher. But it leads to a great gain over the old with full-stroke admission.

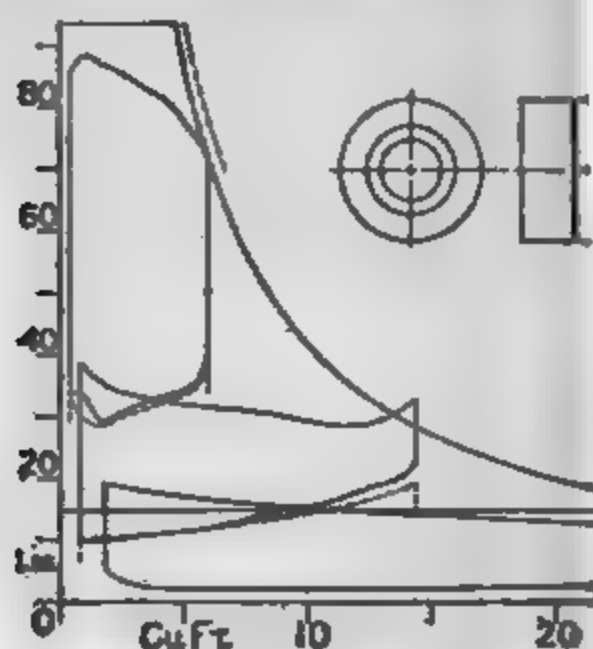


FIG. 600. — Worthington Triple-expansion engine.  
Horizontal, duplex, superheated.  
Test 332, Nov., 1899. E. H. Fox

12	18	29	$\times 18.4$	$N$
			2.25	$H$
.064	.056	.050	5.83	$J$

An efficient example of the latter type of engine. The admission-line of each of the low-pressure cylinders is the exhaust-line of the stage above it. The loss by receiver-drop is, of course, shown in this diagram. The effect of superheating in the very full volumes realized at the end of the receiver-drop loss helping in the analysis of the thermally similar diagrams analyzed in § 36 (h) shows how nearly uniform the results are.

(d) THE EFFECT OF STEAM-JACKS. The diagrams of the Corliss type, and of the various types of driving machinery and electric generators, brought together in Fig. 6

comparison between the systems of non-use and use of steam-jackets—each engine being properly designed for its particular conditions. In A we have a large engine at quite fairly high speed, both size and speed being so great that jacketing would have little if any useful effect: the engine has a reheater in the receiver, but even that was left out of action during the tests. In B, on the other hand, the engine is smaller, the speed is low, and the ratio of expansion is somewhat greater; the jackets are fully applied to barrels and heads of cylinders, and to the receiver. Note that diagram B is drawn for one pound of total steam, the jacket-steam being included under the one-pound saturation-curve, as indicated by the short dotted curve at the top.

To prepare the way for a comparison of thermal actions in these two cases, we note first that engine A is working with a lower admission-pressure and a higher exhaust-pressure than engine B, besides having a less complete expansion; if the two were made alike as to these major conditions, A would slightly surpass B in steam-economy, instead of falling 3.5 per cent. behind, even accepting the larger pressure-loss between the cylinders which is caused by the higher speed. The great advantage of size and speed appears in the greater fullness of the diagram, or the relatively larger volume during expansion, especially in the H.P. cylinder.

In this connection the “quality-curves” plotted in the upper part of the figure are of interest. These give the proportion of the actual steam in the cylinder which is shown by the indicator throughout the expansion: for Case B this would be got by a comparison of the diagram with the abscissa of the dotted steam-weight curve which is only begun on the figure and which would represent the volume of 0.881 lb. of steam, according to the value of  $J$  for this case. Among the numerical quantities under A, the values of  $I$  first given are from the original report, those in parenthesis are from the diagrams as here plotted. The rising of the curves indicates re-evaporation as expansion progresses. Curves of this sort, carefully laid out from accurate measurements, serve to show very clearly the character of the thermal action in the cylinder.

Comparing these two engines, the  
 work in B is to raise  $I$  to about the  
 thus overcoming the influences of low

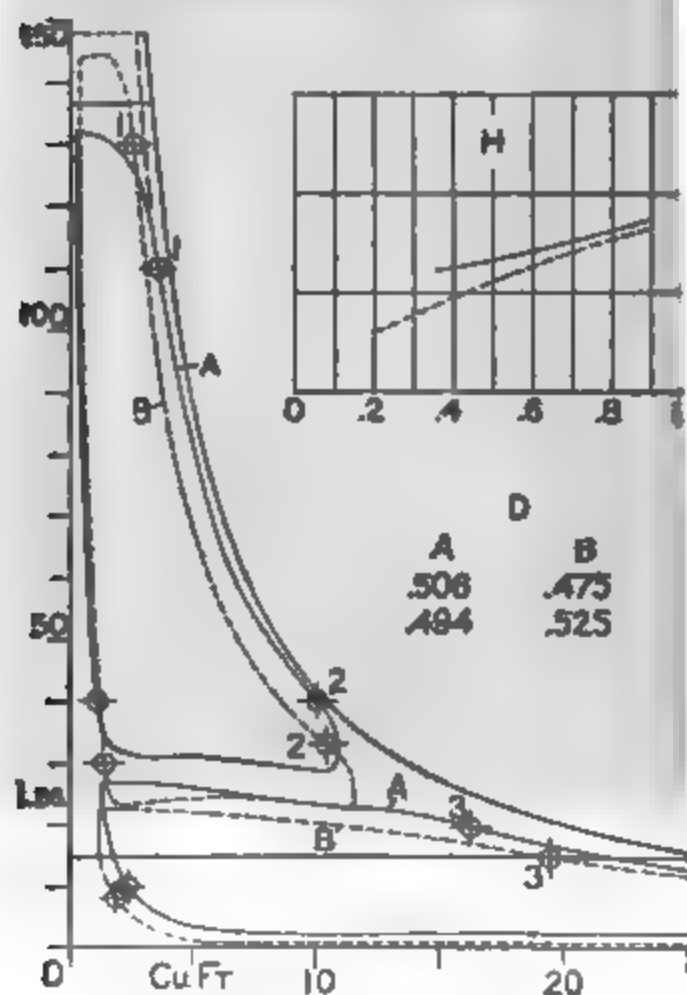


FIG. 601.—Comparison of Unjacketed  
 A. Horizontal, cross-compound, Corli  
 Test 254, 1893. J. E. Denton and ot

30	60	×72		V = 65 2
		3.48		
.026	.036			
				H = 1592
				I <sub>ic</sub> = .875

B Remodelled beam-engine Cortiss ge  
 Test 333, May 1895. Michael Longric

17	31	×60		N = 34 0
1	4	4 085		
0.315	0.27			
				H = 216.
				J = 0.11

but this leaves B nearly 12 per cent.  
 action, so that in effect the jacket-ste  
 cap as to size and speed. Without

would fall a good deal farther behind A in the matter of cylinder-all losses.

Fig. 601 is intended to show typical extreme cases: but a

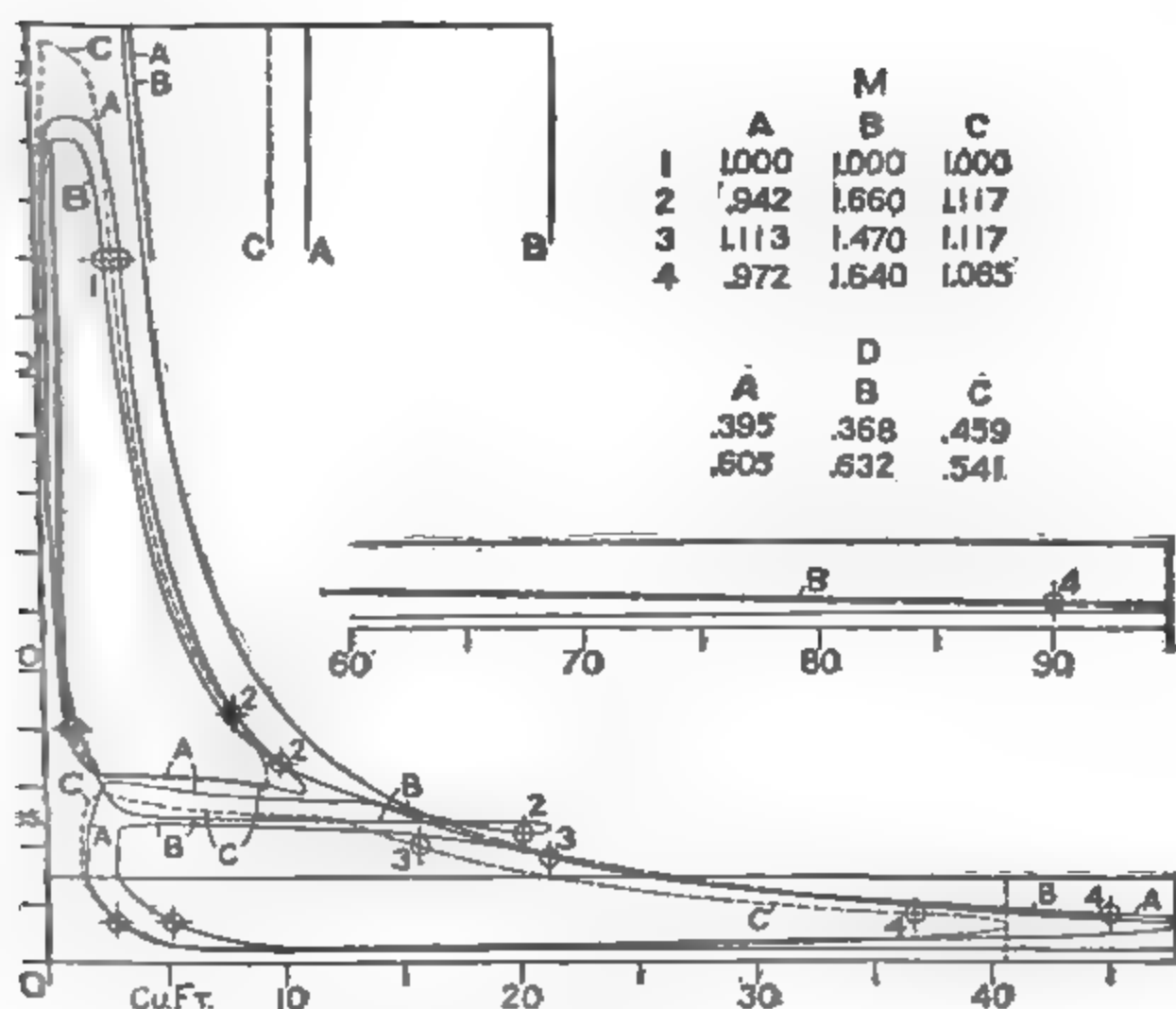


FIG. 602.—McIntosh & Seymour Generator Engine.  
Vertical, cross-compound, slight superheat, reheater, jackets.  
Compare Tests 243, 244; 1901. L. S. Marks, Trans. A.S.M.E., 25-443.

23	48	×48	A, full load, B, half load, with jackets and reheater; C, full load, without jackets and reheater.
5	5	4.40	
.02	.03		

	N	V	$p_m$	H	$S_m$	J	$I_{10}$	$I_{12}$	$I_{15}$	$I_{18}$
A	101.9	815	23.5	1047	13.59	.075	.68	.78	.80	.91
B	103.4	827	12.2	552	13.71	.090	.58	.85	.90	.95
C	101.9	815	24.7	1105	14.41	....	.60	.75	.76	.83

better estimate of jacket-effect can be formed by comparing results from the same engine, as in Fig. 602. For this purpose we consider diagrams A and C. Here the pound of cylinder-steam is the

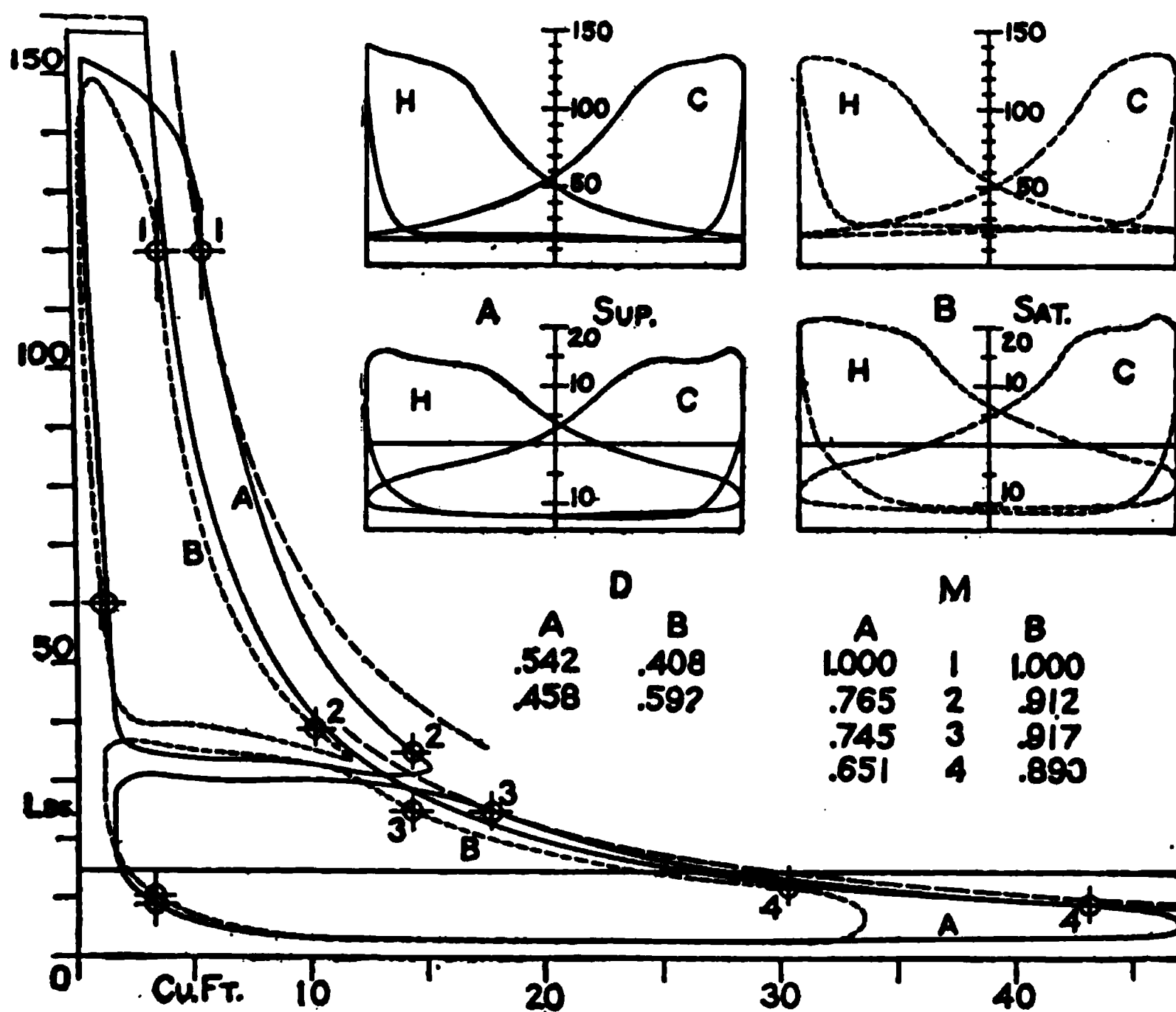
it, which makes the diagram with  $r$  than it really is. The jackets of the H.P. cylinder, but very greatly of the L.P. cylinder: and it is apparent that it holds up the admission-pressure in the L.P. cylinder. The criterion is the value of  $S_H$ , and will be credited with a gain of about 50 per cent. At half load the gain was about ten per cent.

Diagram B, to be compared with Diagram A, shows the effect of condensation and the greater re-expansion in the L.P. cylinder, and also gives an example of a loop diagram. Noting the values under the L.P. cut-off is too early at both loads, and the fact that relatively small condensation in the L.P. cylinder tends to equalise the pressures.

This latter fact, or the change in the main expansion-curve, forms a strong argument in favour of jacketed and unjacketed engines. In the case of the jacketed engine, the steam holds up to and even rises beyond the value of  $S_H$ . In Fig. 602, Case B, the rise is unusually high. In Case C, the rise is unusually low. Case C is anomalous, and is not shown in the original publication, probably because of those here presented. Case A of Fig. 596, with Fig. 596 for smaller cylinders, shows the effect of non-jacketed engines.

(c) THE ENGINE WITH HIGH SUPERHEAT. The effect of the use of highly superheated steam is shown in Fig. 603, with diagrams from the same engine, but with steam dotted in for comparison. The comparison, on the basis of approximately equal steam-weight, shows that the steam (in part of the main supply passes) and the steam is quite highly superheated at H.P. release, and after re-expansion during the first part of the L.P. ex-

superheat, the curves of expansion and compression depart quite widely from the hyperbola: for the H.P. expansion in Case A,



**FIG. 603.—Corliss Engine with Highly Superheated Steam.**  
Horizontal, cross-compound, reheater through which passes a part of the main supply of steam, no jackets.

**Test 291, July, 1903. D. S. Jacobus, Trans. A.S.M.E., 25-264.**

				$N$	$V$	$p_m$	$H$	$S_m$
16	28	$\times 42$	A	102.3	716	31.5	420.4	9.56
3	3½	3.08	B	102.2	715	30.5	406.7	13.84
.041	0.58							

**A with steam superheated about 375°; B with saturated steam.**

the value of the index  $n$  in  $pv^n=C$  is 1.26, found from the coordinates of the points marked 1 and 2 by substituting in the formula

$$n = \frac{\log p_1 - \log p_2}{\log v_2 - \log v_1}, \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (391)$$

which comes directly from  $p_1 v_1^n = p_2 v_2^n$  the expansion-curve drops more rapidly than would the compression-curve rise more rapidly.

This marked change in the shape of the expansion-curve has a marked influence upon the work-division

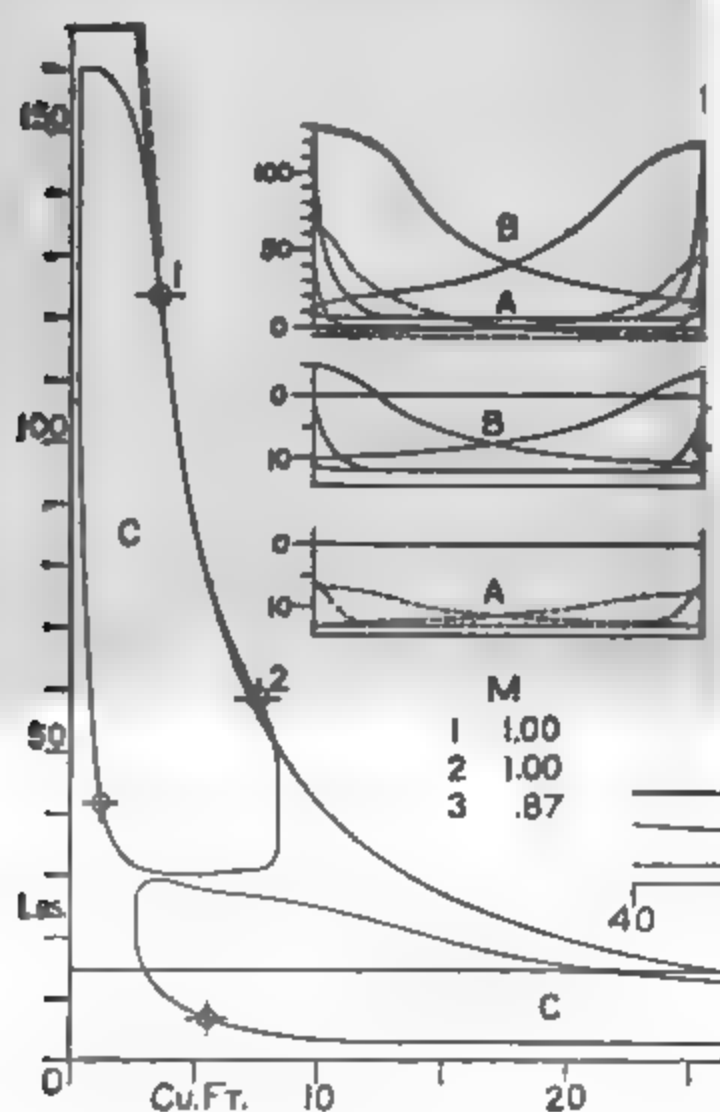


FIG. 604. Fleming Four-valve Engine  
Tandem-compound, reheating rect  
Test 236 2. 1903. B. T. Allen, Tra

15	40	× 27	N = 150 1
24½	24½ - 4½	7 33	H = .
0395	0467		

the L.P. cylinder are relatively so much greater than those of the H.P. cylinder from high superheat that the cylinder-rat same completeness of expansion (as mea



release). The combined diagrams show very clearly that a four to one ratio would be needed for good expansion with saturated steam, with which increase there would be a lowering of the receiver-pressure because of the larger volume at L.P. cut-off. As regards steam-economy, it must be borne in mind that the heat of formation of the pound of steam is very considerably increased when its temperature is raised several hundred degrees.

(f) THE HIGH-RATIO COMPOUND ENGINE.—Typical values of the ratio of the L.P. to the H.P. cylinder in usual practice are 2.4 to 2.8 for non-condensing, 3.5 to 4.5 for condensing engines. Another type of design,\* aiming at full expansion with the fewest possible cylinders, is based on a ratio of about 7, which is a very common total ratio for the triple-expansion engine. The slightly greater thermal losses (as compared with a three-stage engine) are compensated by the fact that there is only one set of receiver losses: but the necessity of more than the desirable amount of receiver-drop, as shown in Fig. 604, partly neutralizes the gain due to greater expansion.

As to the actual performance shown by this figure, it is a little too good to be accepted as entirely reliable: in other words, the smallness of the "missing quantity", as evidenced by the fact that, with only a little clearance-steam back of the diagram, its expansion-curve is thrown out to or beyond the unit-weight curve, does not correspond with the size and speed of the engine and with the use of saturated steam. •

The indicator diagrams illustrate very fully the variation in steam distribution with change of load, the proportions being  $A = \frac{1}{2}$ ,  $B = \frac{1}{3}$ ,  $C = 1$ ,  $D = 1.1$  of rated load. The low admission-pressure in Case B is due to a decrease in the boiler-pressure, while in Case A a further drop in boiler-pressure is supplemented by throttling due to a small valve-opening—see Fig. 493.

(g) TRIPLE-EXPANSION PUMPING ENGINES.—The slow-running pumping engine, with small pressure-losses between the cylinders, and with the thermal losses minimized by the full and effective use

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\* Generally known as the Rockwood system, from Mr. Geo. I. Rockwood, the most prominent introducer.

of jackets and reheaters, gives a higher performance than any other type of engine. Typical diagrams are given in Fig. 605,

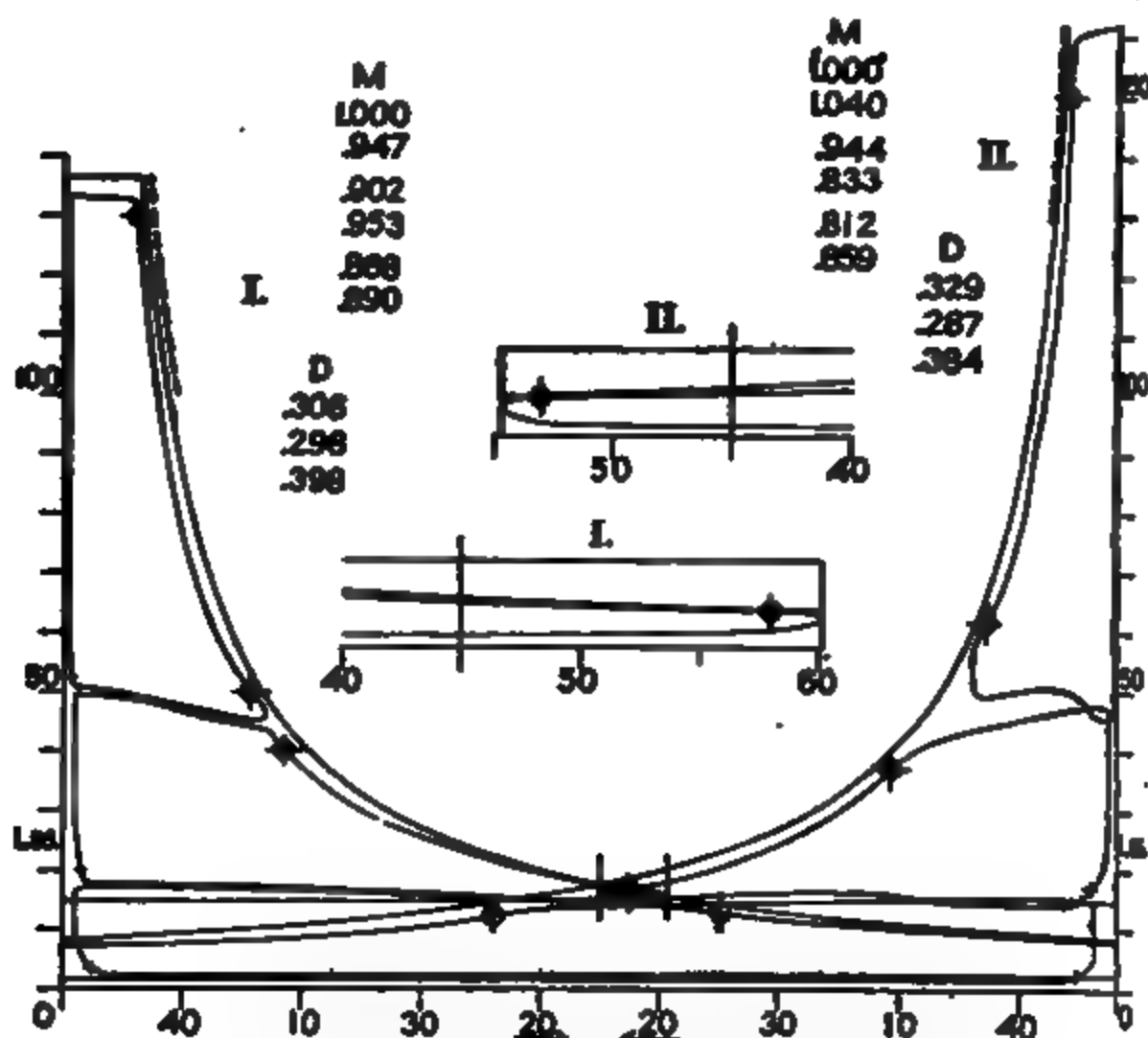


FIG. 605.—Two Vertical Triple-expansion Pumping Engines, reheaters, jackets.

I. Allis-Chalmers Engine at Milwaukee.

Test 353, March, 1893. R. H. Thurston, Trans. A.S.M.E., 15-313.

28	48	74	×60	$N=20.31$	$V=203$	$p_m=26.4$
2-4	2-4	2-4	2.98	$H=573.9$	$S_m=11.80$	
.014	.015	.0077	7.11	$J=0.925$		

II. Snow Engine at Indianapolis.

Test 356, Dec., 1898. W. F. M. Goss, Trans. A.S.M.E., 21-793.

29	52	80	×60	$N=21.18$	$V=212$	$r_m=24.4$
6	7	8	3.26	$H=782.9$	$S_m=11.50$	
.017	.022	.025	7.66	$J=.082$	$I_{ic}=.805$	$I_m=.930$

both from machines of the general form illustrated in Fig. 230. These diagrams call for no comment, their excellence being self-evident.

(h) AN ENGINE WITH A SPECIAL STEAM-CYCLE.—The engine whose performance is represented in Fig. 606 has a number of

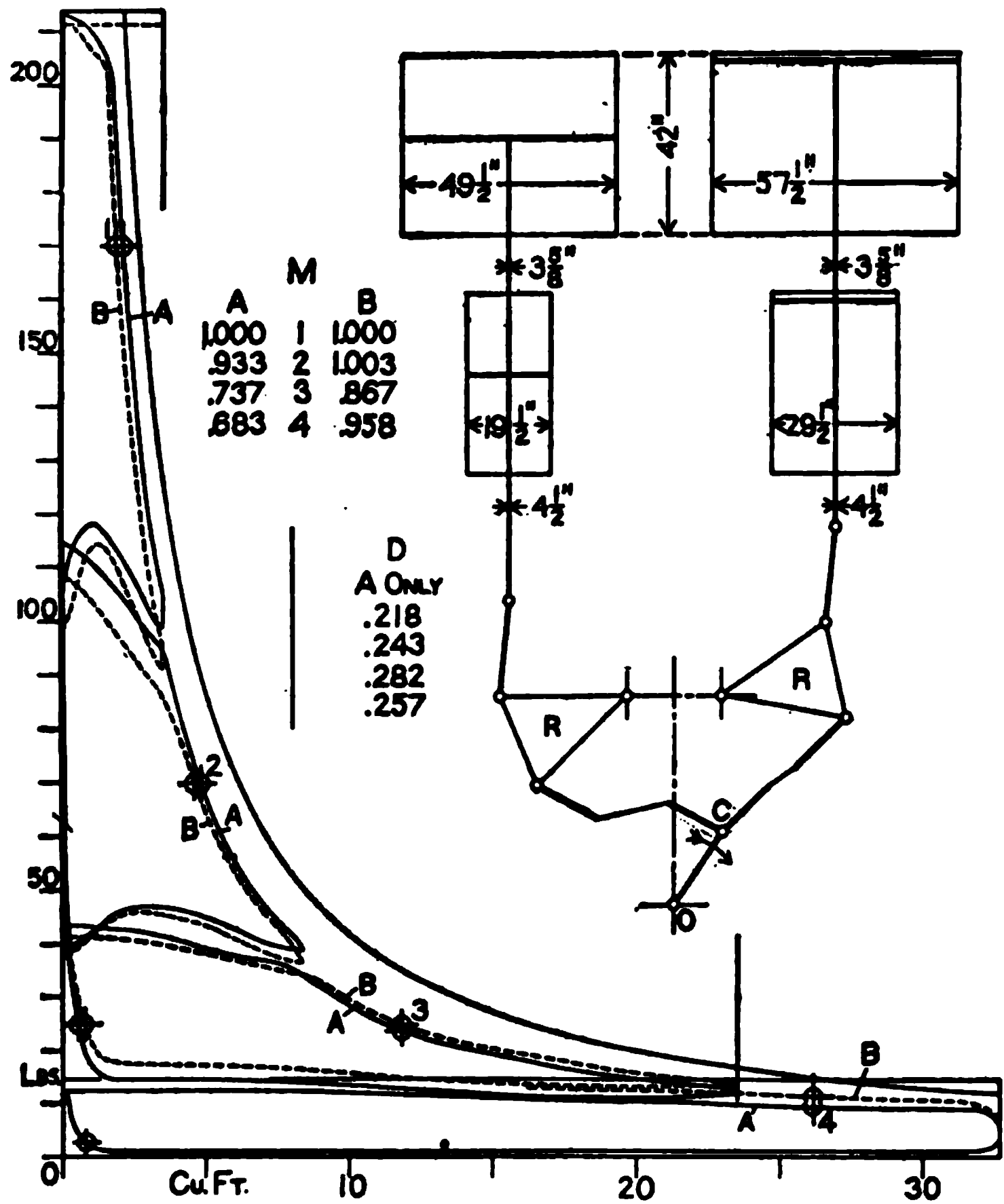


FIG. 606.—Nordberg Quadruple Pumping Engine at Pittsburgh.  
Test 365, 1898. R. C. Carpenter, ENG. REC., April 22, 1899.

Ratios	2.267	6.721	9.076	N=36.52	V=256	p <sub>m</sub> =35.4
Cl.	.0125	.0130	.0055	.0036	H=712.2	S <sub>m</sub> =12.42

points of interest; but the chief is the form of the steam-cycle, which is, in effect, a close approximation to the Carnot cycle.



It will be noted that any jacketed engine in which the jacket-water is either returned at full pressure to the boiler or mixed with the feed, so as to raise the latter above the exhaust-temperature, goes a little way from the Rankine cycle toward the form just described.

On Fig. 606 the diagrams got without the use of the special feed-heaters, but with all the jackets in service, are dotted in. These are laid out for the same power, or on the same base-lines, and are intended to show how the total expansion is modified by the successive abstractions of a part of the working steam. The full-line figures are drawn for one pound of total steam, the jacket consumption not being separately determined.

The mechanism consists of two triangular rockers and of two connecting-rods, with their stroke-lines at right angles, working on one crank. The effect of this, together with the grouping of the cylinders, is as if the four cylinders were in a row, with four cranks, each one at  $90^\circ$  with the one ahead of it. The characteristic action for this arrangement is quite strongly marked in the first two receivers.

(i) TRIPLE-EXPANSION POWER ENGINES, in regard to the amount of clearance and to the form of their diagram in such matters as pressure-loss and throttling effects, lie between the pumping engines as just illustrated and the marine engines which follow; and it has not been thought necessary to give any diagrams from this class. For both factory service and driving generators, triple engines are much more used in Europe than in America, the cost of coal being an important factor in determining whether the slightly more efficient but more costly machine will really be more economical in total charges: see also statements in § 70 (b), under Table 70 A.

(j) MARINE ENGINES.—Not many steam-consumption tests of marine engines have been made, although plenty of indicator cards and of coal-consumption tests are available. The triple engine, in the smaller sizes, is well represented by Fig. 607, both engines having three cylinders, with cranks at  $120^\circ$ . Comparing Fig. 607 with 605, we see that the effective volume at any pressure is in the former, even when account is taken of the fact that

the marine diagrams are drawn for one pound of total steam, the pumping-engine diagrams for one pound of cylinder steam.

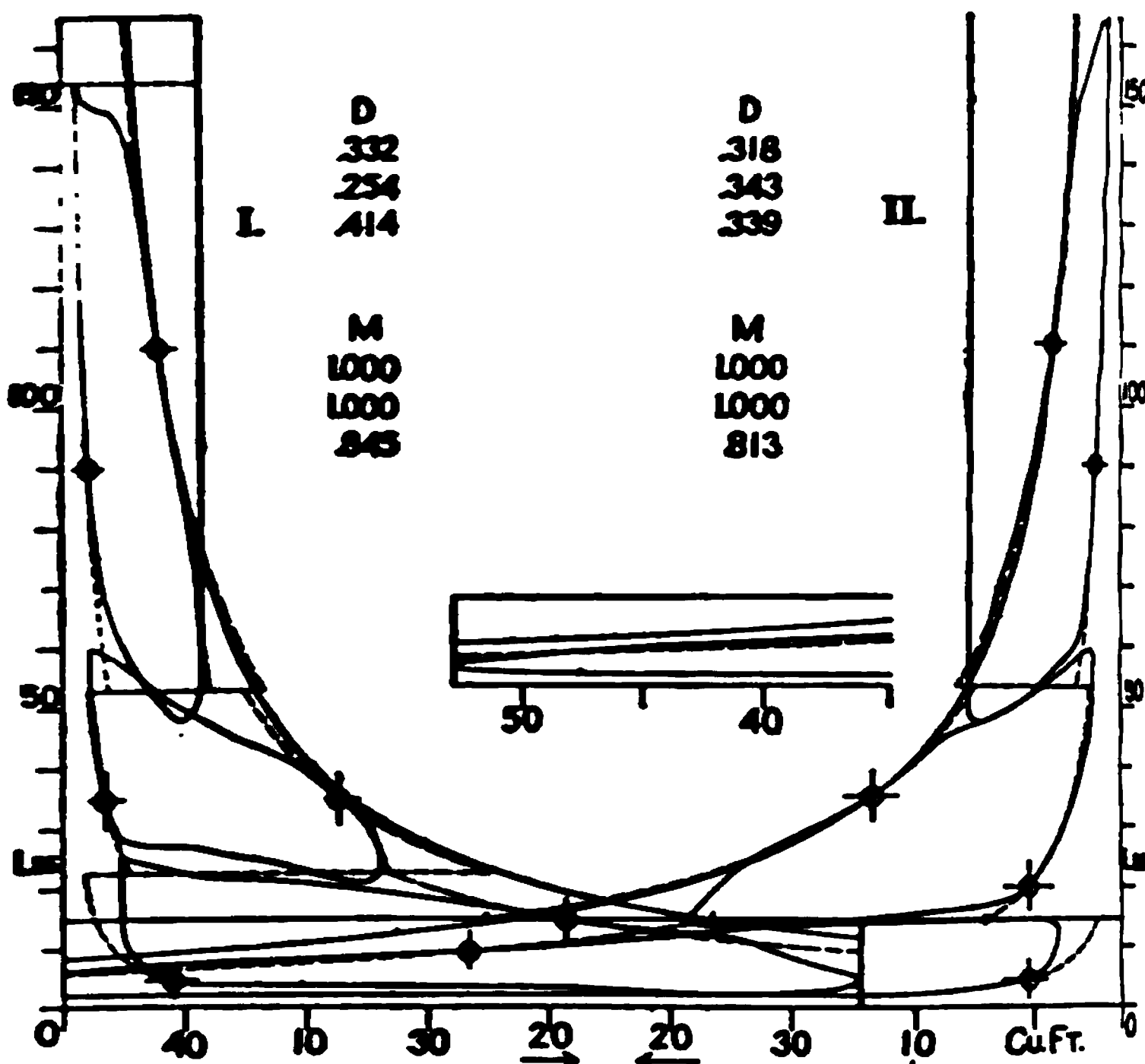


FIG. 607.—Triple-expansion Marine Engines. Tests by Committee of Inst. of Mech. Engs., A. B. W. Kennedy, Chairman.

I. S.S. *Meteor*. Test 382, June, 1888. Eng., 1889, I. 527.

29	44	70	× 48	$N=71.78$	$V=574$	$p_m=29.9$
4½-7	4½-7	4½-7	2.29	$H=1994$	$S_m=14.98$	
.124	.093	.080	5.87	$I_1=.771$	$I_2=.753$	total.

II. S.S. *Iona*. Test 383, July, 1890. Eng., 1891, I. 568.

22	34	57	× 39	$N=61.1$	$V=396$	$p_m=21.1$
5	5	5	2.46	$H=645.4$	$S_m=13.35$	
.124	.101	.076	6.93	$J=.043$	$I=.634, .749, .591$	total.

I. Jackets on all cylinders.

II. Only H.P. cylinder jacketed.

including jacket-feed). This pressure-losses between the cy

mal loss and the larger er with the fact that

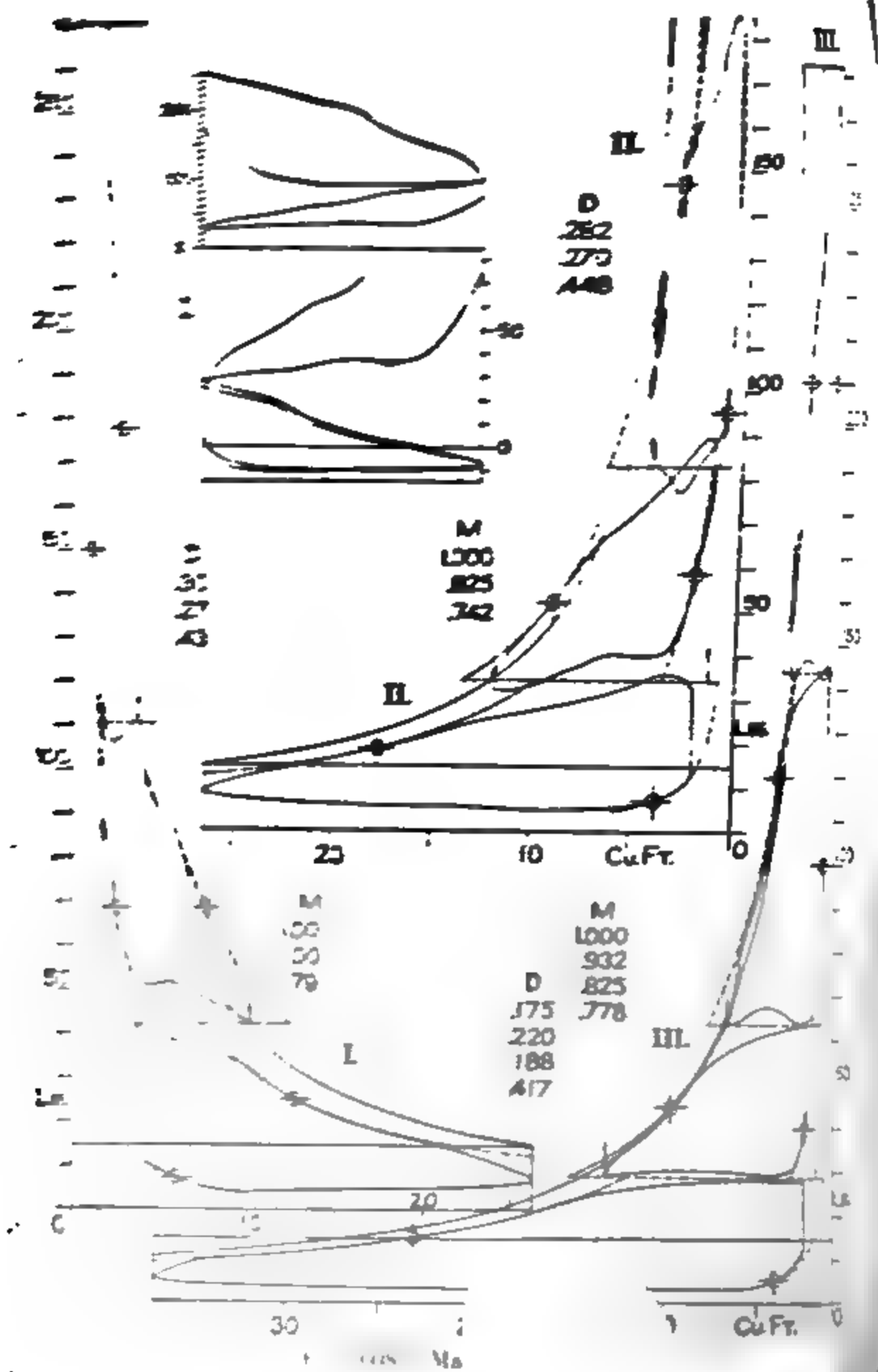
In most cases the expansion is much shorter,\* account for the larger values of  $S_H$  given under Figs. 607 and 608. The *Meteor* engine had jackets on all the cylinders, though their steam-consumption was not separately determined; the *Iona* engine had only the H.P. cylinder jacketed; but in both there is the action characteristic of the non-jacketed engine, as shown by the large drop in the value of  $pv$  in the L.P. cylinder. In spite of its smaller size and somewhat greater thermal losses (apparent on the diagram), the second engine shows better economy because it has smaller receiver-drop losses and fuller expansion. These advantages are due to the earlier cut-offs and to the larger cylinder-ratio.

It will be noted that in the first two stages of both these diagrams the clearance-steam is just about enough in amount to throw the expansion-curves out to the one-pound saturation-curve: in other words, the clearance-steam is nearly equal to the missing quantity of working steam. The diagram from the *Meteor* is redrawn in Fig. 609 with the clearance-steam eliminated, and a rather clearer idea of effective performance can be got from that figure.

The engines in Fig. 608 all have the last stages divided between two L.P. cylinders; I. and II. have four cranks, arranged in pairs opposite each other at the ends, and with these pairs quartered (as in Case B, § 40 (*k*)), but with the angles modified from  $180^\circ$  and  $90^\circ$ ; III. has five cranks. Diagram I. represents the warship engine, which is under requirements similar to those imposed upon the locomotive, in that a large power-development is demanded in proportion to the size of the machine. The striking characteristics are high initial pressure and short expansion or high terminal pressure: and a further noteworthy point is the large gap between stages 2 and 3, with the marked collapse of steam-volume in the last stage. The diagram was made from indicator cards which had been reduced to small size for publication, so that there is room for some error in the scaling; but that this error cannot be very large is shown by the fitting together of the indicator cards in the upper part of the figure, by the close

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\* By short expansion is here meant not a small ratio of expansion, but a small absolute volume of the pound of steam when it is exhausted.





agreement of the hyperbola with the L.P. expansion-curve, and by the confirmatory results in diagram II. It is hard to see, however, why there should be so great a drop of pressure between these two stages. The two L.P. indicator cards, each a mean for its cylinder, are sketched together to show how nearly they agree: in the combined diagram their average is used, the whole diagram being drawn as the mean for the two engines, or from four sets of cards. The engines have jackets, but they were not in service when these diagrams were taken. In general, the economy was better without the jackets.

In Fig. 608 II. the diagrams are accurate, but the steam-consumption is only assumed at a probable value: the volume-scale corresponds with 15 lbs. per horse-power-hour, and the resulting position of the saturation-curve is about what should be expected in view of the size and speed of the engines. These have jackets, but they are not usually filled with steam when the engine is running, being rather used for warming the cylinders before starting. As in engine I., there is a decidedly poor vacuum realized in the cylinder: but with such short expansion high vacuum does not add much to the performance of the engine.

Fig. 608 III. shows an action in which there is evidence of a strong and successful effort after economy. The steam is superheated about 80 degrees, besides being made at a very high pressure;

FIG. 608—(Continued).

I. H.M. Cruiser *Argonaut*. Sir John Durston, paper before Inst. Nav. Arch. Test 385. ENG., 1899, I. 391, 432.

Two	34	55	64-64	×48		$N=115.3$	$V=922$	$p_m=38.3$
	$5\frac{1}{2}$	$5\frac{1}{2}$	$5\frac{1}{2}$	2.685		$H=13,788$	$S_H=15.44$	
	.24	.20	.16	7.155				

II. S.S. *Kaiser Wilhelm der Grosse*. ENG., 1898, I. 649.

Two	51.9	89.8	96.5-96.5	×68.9	2.98		$N=78$	$V=896$
	.16	.12	.08		6.98		$p_m=37.4$	$H=29,150$

III. S.S. *Inchdune*, five-cylinder quadruple. ENG., 1901, I. 71.

	17	24	34	42-42	×42		$N=73$	$V=511$
Cl. . . .	.20	.13	.11	.06			$p_m=35.2$	$H=1627$
Ratios	2.00	4.00	6.12					

Steam-consumption not determined in last two; estimated at 15 lbs. in II., at 11 lbs. in III.

the cylinders are fully jacketed, and the expansion is quite full; and the data as to economy is the statement of 0.97 lb. of coal per H.P.H. This applies but the assumption of 11 lbs. of steam out of the way.

(k) THE DISTRIBUTION OF THE LOAD engine is well illustrated by Fig. 609, the idea of collecting into concise form the preceding discussion. Diagrams Fig. I., are redrawn with the compression-axis, so as to eliminate the clean saturation or constant-weight curve Fig. as a reference-line in all the combinations been given. Curve B is the adiabatic dry steam at boiler-pressure, and running the same as that which the actual steam have if produced to the end of its discharge curve is drawn a hyperbola, marked Then losses due to shrinkage of volume condensation (with leakage included) are by one direction of cross-hatching, losses free expansion are grouped together cross-hatching. There will be some division will appear more or less arbitrary being open to dispute. The idea Fig. F is that if the full adiabatic volume were fill a larger space when at the exhaust, really the proper reference-curve when dynamic performance, but for ordinary curve is rather more useful, since it enters of the steam in the cylinder.

(l) THE DIAGRAM FACTOR.—For the approximate size of the L.P. cylinder to develop a certain power, it is convertible roughly, by a simple computation, with  $P_m$  is likely to be. One way is to

engines, of the "coefficient of fullness" or diagram factor, this being the ratio of the effective area of the combined diagrams to a simple circumscribing diagram with the  $(pv=C)$ -line for its expansion-curve.

One method is to take the ordinary combined diagram, with the clearance-line as vertical axis, and draw the hyperbola to touch

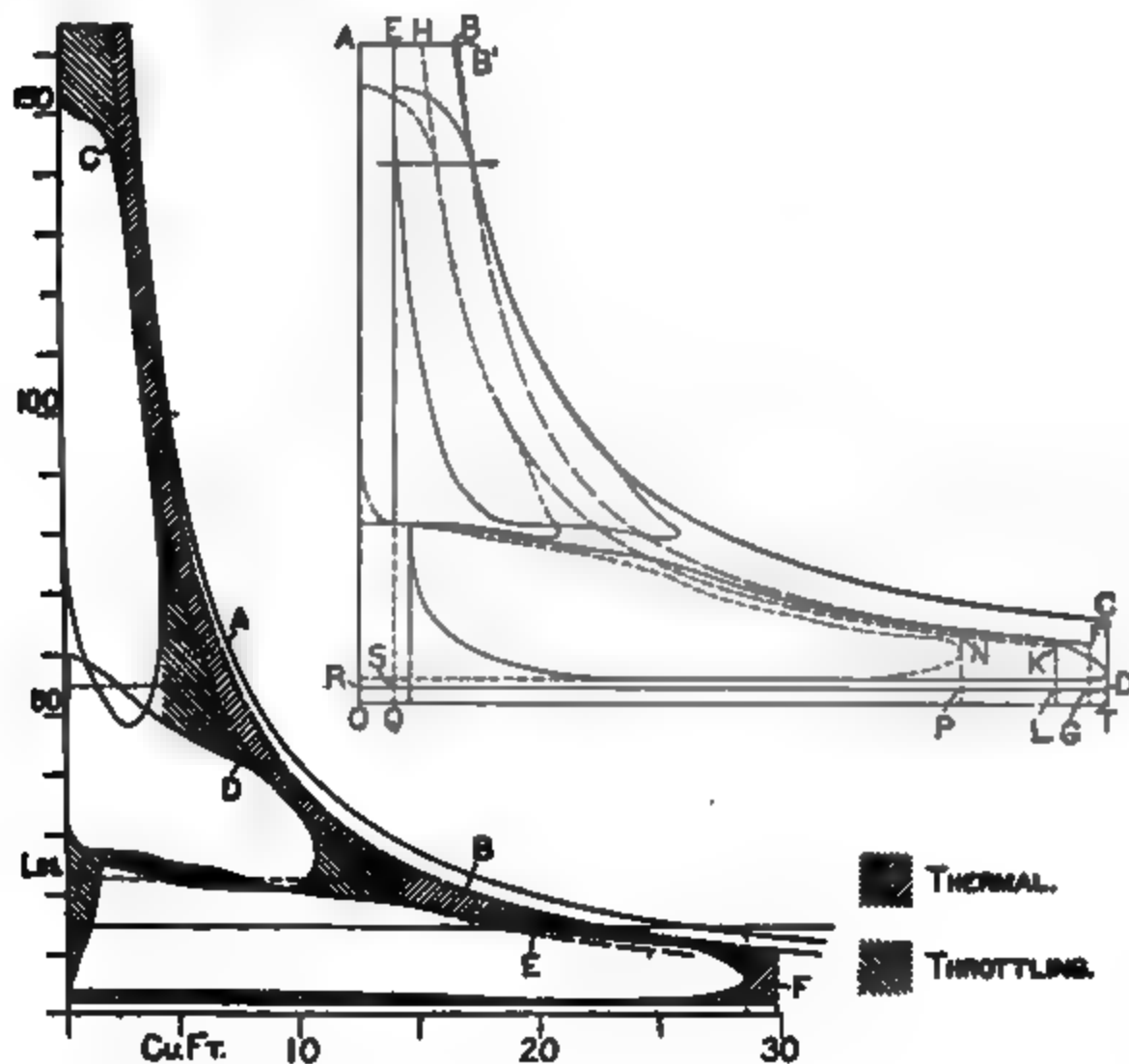


FIG. 609.—Diagrams from S.S. *Meteor*, referred to compression-curves.

FIG. 610.—To Illustrate the Diagram Factor.

the H.P. expansion-curve at cut-off, extending it from the boiler-pressure to the end of the L.P. diagram. In Fig. 610 this makes ABCDR the ideal figure, though some engineers have used the base-line OT instead of the condenser-pressure line RD as the bottom of the diagram. Using RD, the area of the indicator is 57.2 per cent. of the area of ABCDR. Busley, in his

the *Marine Steam-engine* (original in factors which lie in the neighborhood 0 to 0.75 for various classes of machines.

The really logical scheme is to transfer the dotted position, and make the common axis AO (as in Fig. 609); then the high-pressure as origin, and the effective combined area coming here to 84.7 per cent. But this work and changes the lengths of the thing else than the actual cylinder-volume.

For convenient practical use the base of the several end-lines to a common axis. The L.P. diagram were moved a little way to touch EQ. Then with Q as origin to touch the H.P. expansion-curve line through the upper end of the common diagram QEB'F have the same OAHN. The length SG is the same representing, as is proper, the volume of this area EB'FGS as standard, the factor of the mean pressure of any of these similar diagrams be calculated as in § 17 (c), with Tabular work involved.

Variation in the standard used by the diagram-factor method less useful than the roughest preliminary work it is better to measure it.

(m) THE SINGLE-ACTING TANDEM-COMPUND. To complete this subject of compound-engines, note of the peculiar action which is characteristic of an acting compound engine is changed from the Willans form (Fig. 255) to the Willans form (Fig. 256) variation in the pressure on the under-surface of the piston it is natural to use the variable space between the next cylinder-head as the receiver; but in the receiver acts on a moving surface, the indicator card from the receiver, and the

the "wrong side" of the piston. A typical combined diagram from a two-stage Willans engine is reproduced in Fig. 611. Referring to Fig. 439, we see that on the up- or exhaust-stroke the combined volume of cylinder  $C_1$  and receiver  $R$  is constant, so that, barring secondary influences, the steam-pressure throughout this space will be constant, as shown by the horizontal line between diagrams  $C_1$  and  $R$  in Fig. 611. During the down-stroke the receiver pressure drops till L.P. cut-off, then is raised by compression: so that on the lower side of the H.P. piston is done the work represented by the area of diagram  $R$ . In any ordinary direct-expansion engine, with receiver of constant volume, the separate areas  $C_1$  and  $R$  are combined in a single figure.

Somewhat more complex in its action is the Schmidt "motor" represented in Fig. 612. On the down-stroke, steam is admitted

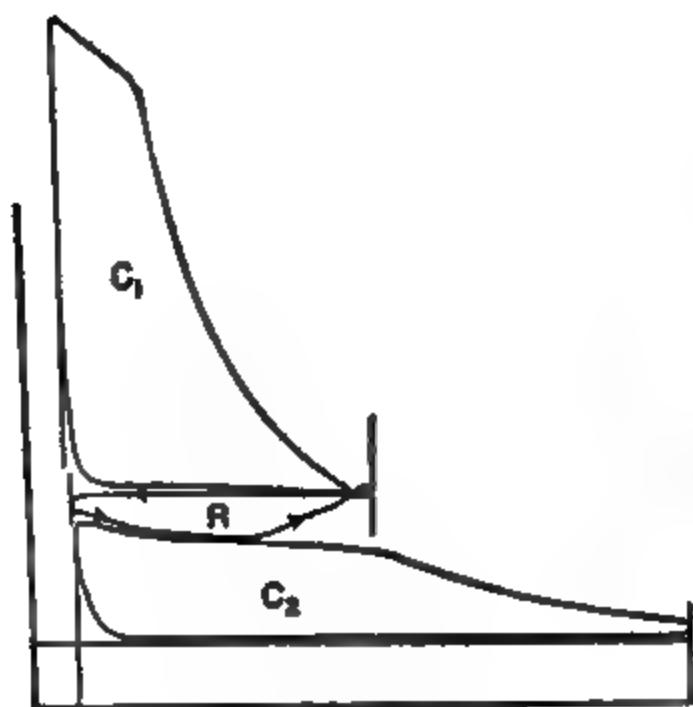


FIG. 611.—Diagrams from Compound Willans Engine.

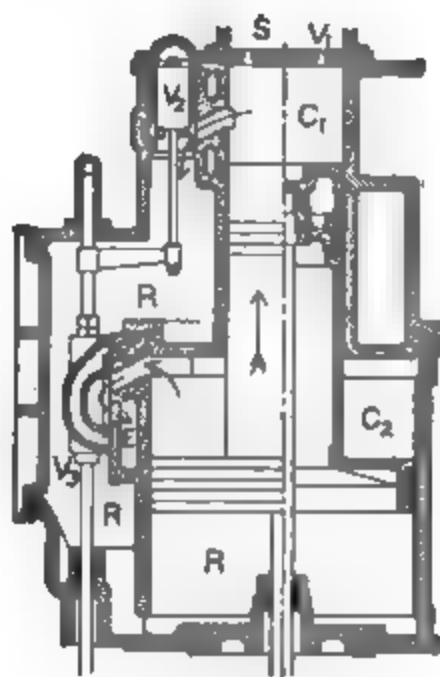


FIG. 612.—Section of Schmidt "Motor" for Superheated Steam.

to  $C_1$  through the annular port in the cylinder-head, controlled by a lift valve  $V_1$ , and  $V_2$  is open, admitting steam to  $C_2$ , and equalizing the pressure in  $R$  and  $C_2$ ; but the communicating volume decreases, because the H.P. piston advances into it, and the receiver-pressure rises. On the up-stroke  $C_1$  is open to  $R$ ,

*The Marine Steam-engine* (original in German), gives a collection of factors which lie in the neighborhood of 0.67, ranging from 0.60 to 0.75 for various classes of marine engines.

The really logical scheme is to transfer the diagrams into dotted position, and make the compression-curves agree with axis AO (as in Fig. 609); then the hyperbola HN is drawn with A as origin, and the effective combined area is compared with AHN, coming here to 84.7 per cent. But this involves too much graphical work and changes the lengths of the respective diagrams to something else than the actual cylinder-volumes (as laid off to scale).

For convenient practical use the best method is simply to transfer the several end-lines to a common axis, as would be done if the L.P. diagram were moved a little way to the left in Fig. 610 so as to touch EQ. Then with Q as origin the hyperbola BF is drawn to touch the H.P. expansion-curve at a point on a horizontal line through the upper end of the compression-curve—this makes the diagram QEB'F have the same width, at any pressure, as OAHN. The length SG is the same as that of the L.P. diagram, representing, as is proper, the volume of the large cylinder. With this area EB'FGS as standard, the factor is 0.806. The area of the mean pressure of any of these simple ideal diagrams can easily be calculated as in § 17 (c), with Table I. to diminish the arithmetical work involved.

Variation in the standard used by different writers makes the diagram-factor method less useful than it might be. Even for the roughest preliminary work it is better to sketch out the diagram and measure it.

(m) THE SINGLE-ACTING TANDEM-COMPOUND ENGINE.—To complete this subject of compound-engine diagrams, we must note of the peculiar action which is introduced when a single-acting compound engine is changed from the Westinghouse form (Fig. 255) to the Willans form (Fig. 439). To prevent any variation in the pressure on the under side of the higher piston it is natural to give a variable space between this piston and the next cylinder below the receiver; but since the pressure in the receiver is a moving quantity, the indicator cannot be received.

"wrong side" of the piston. A typical compound diagram from a two-stage Willans engine is reproduced in Fig. 611. Referring to Fig. 439, we see that on the up- or exhaust-stroke the communicating line of cylinder  $C_1$  and receiver  $R$  is constant so that, having no secondary influences, the steam-pressure throughout this space will be constant, as shown by the horizontal line between diagrams  $C_1$  and  $R$  in Fig. 611. During the down-stroke the receiver pressure drops till L.P. cut-off, then is raised by compression so that the lower side of the H.P. piston is now the work represented by the area of diagram  $R$ . In any ordinary direct-expansion engine, with receiver of constant volume, the separate areas  $C_1$  and  $R$  are combined in a single figure.

Somewhat more complex in its action is the Schmidt "motor" presented in Fig. 612. On the down-stroke, steam is admitted

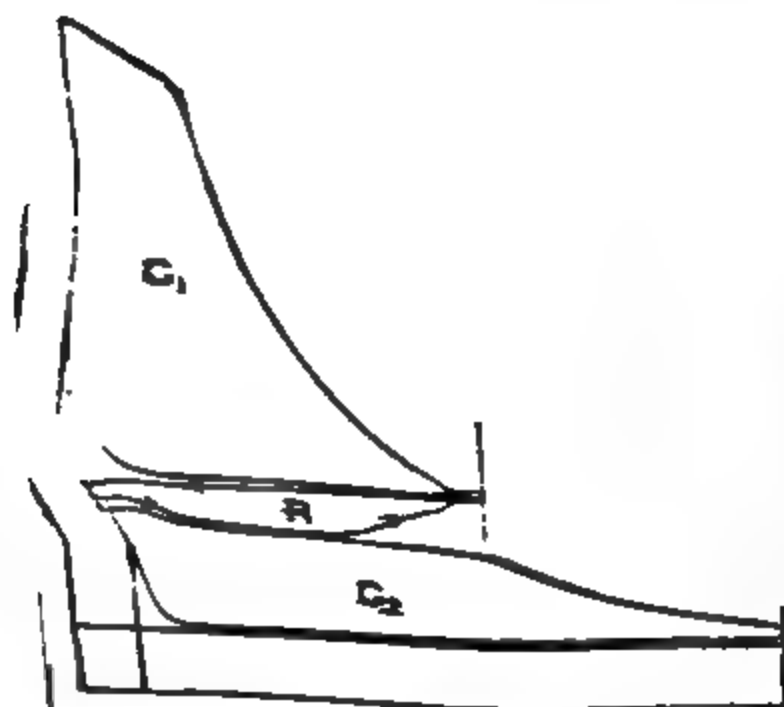


FIG. 611.—Diagram from Compound Willans Engine.

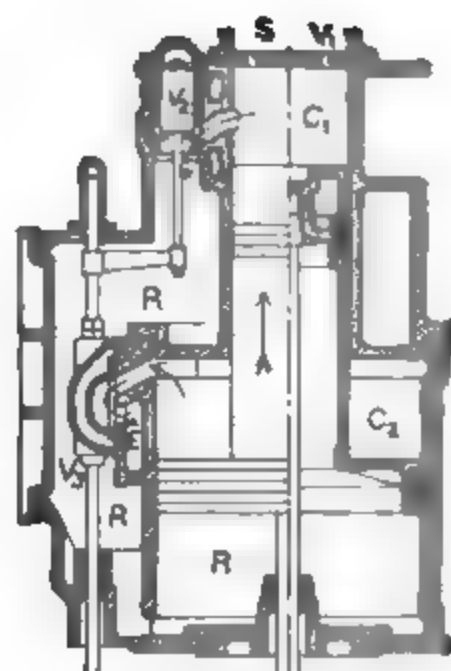


FIG. 612.—Section of Schmidt "Motor" for Superheated Steam.

to  $C_1$  through the similar port in the cylinder-head, controlled by a lift valve  $V_1$  and  $V_2$  is open, admitting steam to  $C_1$ , and  $C_2$ ; but the communicating I.P. piston advances into it, and the up-stroke  $C_1$  is open to  $R$ ,

and the receiver-pressure falls on acc to the displacement of the annular & the small one. Still, the receiver-pr exhaust-pressure, so that there is a proximately equal to that in the de what are essentially single-acting p effect of the receiver gets the force-pl The M.E.P. of the receiver diagram upon the full bottom area of the p or power: but when the diagrams ar to "reduce" this diagram to a short increasing the ordinates as the base-li by Professor Doerfel in *Zeitschrift des* 1899, page 1562. The result looks ve

### § 70. Cylinder Proportio

(a) RANGE OF DATA.—Numerous & dimensions will be found scattered forth under the diagrams in the 1 gathered together in the Test Table these, and for the better presentatio subject, the tables in this section ha

The two-stage engine is so fully ar 75 B and C that very little need be ad first group, Nos. 1 to 7, is made up of en ratios; while Nos. 8 to 10 are intende portions in compound marine engine to 374. Table 70 B gives a set of multiple-expansion power and pump values of the total ratio in the first and repeated from Tables 75 C and D, bu fuller analysis of the relations between field of marine engineering is sufficient

(b) THE COMPOUND ENGINE.—The writer has knowledge is found in some for use on British warships, with cylinde being 1.56. but while working on a



this engine exhausts at high pressure into the second receiver of the main engine or into the evaporator. For engines with normal exhaust,\* the ratio 2.0 is about the limit. Example No. 1, Table 70 A, is an extreme case for a non-reversing engine. If the engine is to start frequently, as Nos. 2 and 3, the use of a very large H.P. cylinder is desirable, because a strong initial torque is thereby secured. Compound winding (hoisting) engines are not very common, but the use of two stages in a reversing rolling-mill engine (No. 3) is decidedly unusual. The designation 2-crank or 3-crank in these cases means that there are two or three tandem-compound units.

The compound locomotive, running always non-condensing and often with a back-pressure well above atmosphere, has a low cylinder-ratio, in spite of its high boiler-pressure. The example in Table 75 B vary from 2.35 to 2.8, and cover the range of practice very fairly. The small stationary high-speed engine, represented by Figs. 202, 203, 206, and 208, has about the same range: but if the engine is to be run condensing at times a slightly higher ratio is desirable, as in Test 216. An unusually low ratio is seen in Fig. 612, and the same condition is indicated by the high values of  $p_m$  in Tests 224 and 225.

For the larger condensing engine we see one line of practice in Nos. 4 to 7 of Table 70 A and in Tests 281 to 285, with the ratio from 2.3 to 3.0; another line in Tests 241-4 and 251-9, where the ratio ranges from 3.3 to 4.5. The evident belief that the individual cylinder-ratios must be small will account for the much more general use of the triple-expansion engine for ordinary power service on the Continent of Europe than in England and especially in America. With us the idea has been rather to get the full amount of expansion out of each cylinder and to use fewer stages.

This idea receives its fullest development in the Rockwood type of high-ratio compound, already commented upon in § 69 (f), and with the performance set forth in Tests 271-4. This scheme runs into the triple engine field as to proportions, and under comparable conditions gives nearly if not quite as good economy.

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\* The engines in the tables here given are all to be run condensing except No 3

TABLE No. 3. Comparison of Various Designs

No.	MAKER REFERENCE	TYPE SERVICE	Dimensions Feet	Weight Tons	Speed Knots	Cost \$
1	DUIJBERGER MACH. CO. ENG. 1902, II. 537	Hor. tand. lift valve	20 0 41 0	0 100	1 100	1000
2	GUTHRIE & SONS LTD. ENG. 1902, I. 813	Roll mill motor	4 0 0	0 100	1 100	1000
3	COCKERILL, GHEENT ENG. 1905, II. 376	Hor. 2-ck. tand. lift Motor house	30 0 41 0	0 100	1 100	1000
4	CARELS FRÈRES, GHEENT ENG. 1900, II. 79	Hor. 3-ck. tand. lift	30 0 41 0	0 100	1 100	1000
5	BOLLINCKX, BRUSSELS ENG. 1900, II. 368	Roll. mill motor	30 0 41 0	0 100	1 100	1000
6	ESCHER, WYSS & CO. ENG. 1900, II. 296	Hor. tand. lift valve (Generator)	30 0 41 0	0 100	1 100	1000
7	GREVENBOICH ENG. 1903, I. 40	Hor. 2-ck. (Corliss) (Generator)	30 0 41 0	0 100	1 100	1000
8	DENNY, DUMARTON ENG. 1891, I. 40	Hor. tand. (Corliss) (Generator)	30 0 41 0	0 100	1 100	1000
9	ANBALDO, GENOA ENG. 1896, I. 342	Hor. 2-ck. tand. lift (Generator)	30 0 41 0	0 100	1 100	1000
10	MARYLAND STEEL CO. MAR. ENG., Jan. 1905	Inclined 2-crunk Paddle steamer	30 0 41 0	0 100	1 100	1000
		Vert. 4-ck. (duplex)	47 0 0	0 100	1 100	1000
		Battleship, 2 screws	47 0 0	0 100	1 100	1000
		Vert. 4-ck. (duplex)	30 0 41 0	0 100	1 100	1000
		Ferryboat, 1 eng.	30 0 41 0	0 100	1 100	1000

REFERENCES - in this and the following tables:

Author	Reference	Page
A. S. M. E.	Trans. Am. Soc. M. E., Vol. 1891	100
ENG. REC.	Engineering, London	100
MAR. ENG.	Engineering Record, New York	100
Z. V. D. I.	Zeitschrift des Vereins deutscher Ingenieure, New York	100

TABLE 70 B. MULTIPLE-EXPANSION ENGINES.

No.	TYPE REFERENCE	DIAMETERS SERVICE STROKE	RATIOS	$P_{IG}$ $P_m$	N	H	V
11	Vert. 3-ck. lift-valve ENG. 1902, II. 179	32.7, 55.1, 80.7 Generator	2.83 6.10	171	94	740	
12	Vert. 2-ck. lift-valve Z. V. D. I. 1900, 606	34.0, 49.1, 61.0-61.0 Generator	2.08 6.40 (T 316)	26.2 200	3000 83	700	
13	Vert. 2-ck. Corliss ENG. 1896, I. 129	11.5, 19, 30 Generator	2.73 6.79 (T 302)	25.0 120	3000 75	450	
14	Vert. 4-ck. Corliss ENG. 1905, I. 539	37, 59, 72-72 Generator	2.54 7.58	26.5 190	225 75	750	
15	Hor. 2-ck. Corliss ENG. 1893, II. 694	26, 40, 60, 70 Generator	2.37, 5.32, 7.25 2.25, 1.36	32.0 180	6000 60	720	
				35.9	3000		
16	Vert. 3-ck. Cor. and lift ENG. REC., Dec. 2, 1893	28, 48, 74 Pump. eng.	3.15 6.99 (T 353)	120 22	20 575	200	
17	Vert. 3-ck. Cor. and lift ENG. REC., Nov. 16, 1901	22, 41.5, 62 Pump. eng.	3.56 7.95 (T 357)	150 21	25 475	250	
18	Vert. 3-ck. Cor. and lift ENG. REC., Oct. 13, 1900	30, 56, 87 Pump. eng.	3.48 8.40 (T 358)	190 23	18 800	200	
19	See Fig. 606 (T 365) ENG. REC., Apr. 29, 1899	19.5, 29, 49.5, 57.5 Pump. eng.	2.21, 6.44, 8.70 2.92, 1.35	200 35	36 700	250	
20	Horiz. 4-ck. (T 366) A. S. M. E., 28 -	14.5, 22, 38, 54 Air-comp.	2.30, 6.86, 13.92 2.98, 2.03	240 31	57 1000	460	

RATIOS:

First line = ratio of each successive cylinder (or stage) to the first or high-pressure cylinder.  
Second line = ratio of each cylinder (after the second) to the one ahead of it.

These are all for the nominal volumes or piston-displacements (not considering the clearances), and do not include the effect of the piston-rods.

Numbers like (T 316), etc., refer to Test Tables in Chapter XIII.

TABLE 70 C. MARINE ENGINES.

No.	VESSEL REFERENCE	ENGINES	DIAMETERS STROKES	RATIOS	Ratio	$P_{10}$ $P_0$	N	H	V
21	INDIANA Fig. 234 U. S. B. S.	34.5, Two	48, 75	1.93	4.73	150	131	917	
22	MINNEAPOLIS U. S. Cr. Eng. 1894, II. 563	42, Three	42, 92	1.98	2.44	38.9	9500		
23	MAGNIFICENT Br. B. S. Eng. 1895, I. 485	40, Two	59, 88	2.18	4.86	150	132	925	
24	OCEAN Br. B. S. Eng. 1899, II. 152	30, Two	51, 80	2.67	2.22	55.0	20,400		
25	GOOD HOPE Br. B. S. Eng. 1902, I. 286	43.5, Two	71, 81.5-81.5	2.67	2.66	155	12,000		
26	DEVONSHIRE Br. Cr. Eng. 1905, I. 550	41.5, Two	73.5-73.5	2.49	7.04	250	108	918	
27	OHIO U. S. B. S. Mar. Eng., Sept., 1904	35.5, Two	53, 63-63	2.23	2.64	48.4	13,500		
28	VIRGINIA U. S. B. S. Mar. Eng., June, 1904	35, Two	57, 66-66	2.65	6.27	278	125	1000	
29	MARYLAND U. S. Cr. Mar. Eng., March, 1905	38.5, Two	63.5, 74-74	2.72	2.52	49.1	31,000		
30	U. S. SCOUT CRUIZERS Mar. Eng., June, 1905	28, Two	45, 62-62	2.49	2.82	200	140	980	
			36	3.80	2.68	42.7	21,500		
					7.11	210	124	992	
					2.82	42.8	16,000		
					2.68	250	120	960	
					2.72	47.9	19,000		
					7.30	250	128.5	1028	
					2.72	51.5	27,800		
					9.63	250	200	1200	
					3.80	36.5	16,000		

TABLE 70 C—Continued.

No.	VESSEL REFERENCE	ENGINES DIAMETERS STROKE	RATIOS	$P_{10}$ $P_m$	N H	V
31	NEW YORK 1887 ENG., 1888, II. 123	45, 71, 113 Two	2.48 6.29	150	20,000	
32	NEW YORK 1902 MAR. ENG., May, 1903	42, 66.5, 77-77 Two	2.51 6.72		95 950	
33	CAMPANIA 1892 ENG. 1893, I. 480	37-37, 79, 98-98 Two	2.28 7.02	37.4	20,000	
34	ST. LOUIS 1894 ENG. 1895, I. 800	28.5 } 55, 77, { 77 28.5 } 60 { 77	1.86, 3.65, 7.30 1.96, 2.00	165 84 966 34.0 30,000	200 95 950 37.4 20,000	
35	KR. WILHELM D. GR. 1897 ENG. 1898, I. 429	52, 89.7, 96.4-96.4 Two	2.97 6.86	178 78 893		
36	OCEANIC 1899 ENG. 1899, I. 53	47.5, 79, 93-93 Two	2.76 7.66	38.0 30,000		
37	DEUTSCHLAND 1900 ENG. 1900, I., II.	36.6 } 73.6, 103.9 { 106.3 36.6 } 72.8 { 106.3	2.05, 3.91, 8.52 1.91, 2.18	220 27,000	77 935	
38	IVERNIA 1900 ENG. 1900, II. 436	28, 41, 58½, 84 Two	2.14, 4.33, 9.00 2.02, 2.08	36.2 36,000		
39	KABER WILHELM II. 1902 ENG. 1903, II. 37-	37.4, 49.2, 74.8, 112.2 Four	1.71, 4.01, 9.00 2.35, 2.24	225 80 945	40,000	
40	MINNESOTA 1902 MAR. ENG., May, 1903	29, 51, 89 Two	3.08 9.42	230 78 702		
			3.06	37.8	10,000	

(c) THE MULTIPLE-EXPANSION ENGINE. In the case of the engines B and C, with the latter part of 75 C, it is seen that of all the natural and obvious increases in the cylinder-ratio, the rising steam-pressure, and also the temperature, the American and English than in Germany appears that the usual range in model engines is with 9 as an advanced proportion. This is decidedly a special case. In regard to the system of working described in the case of the engine, that, on account of the diminution of steam-pressure during expansion, the cylinder-ratio is relatively even, and the same values would indicate.

In considering the matter of the cylinder-ratios, we wish to see, roughly at least, how the simple theory are modified by secondary influences. The last stage has relatively a much larger cylinder-ratio at the end of expansion to the exhaust than the other stages: consequently, this cylinder-ratio is cut-off, and a smaller ratio to its upper limit is shown up very clearly in the pumping cylinder, where the secondary influences are small. In No. 11; it appears also in the quadruple expansion. But in the marine engines, on account of the small clearance and of intermediate losses, an equal cylinder-ratio requirement of nearly the same cut-off is necessary. In a progressive increase in the proportion of expansion, the ratios become practically equal, 34, 38, and 40; while the peculiar condition of the normal manner of progression is seen in the case of the engine.

When a stage in the expansion is divided into two, of which each is a work-unit, then that stage is equal to the total volume. The usual case is the case of the engine, here the low-pressure ratio should be taken as 14, 27, 30, and 32; when it is equal, as in the case of the engine 36 (all recent designs), we see the evident influence of the secondary influences were perhaps not fully taken into account in the design. A close and extensive

diagrams would be necessary to make clear just what these influences are, going beyond the scope of the present discussion. Note that No. 13 (see Fig. 613 III.) is essentially the same as a four-cylinder triple in regard to work-division.

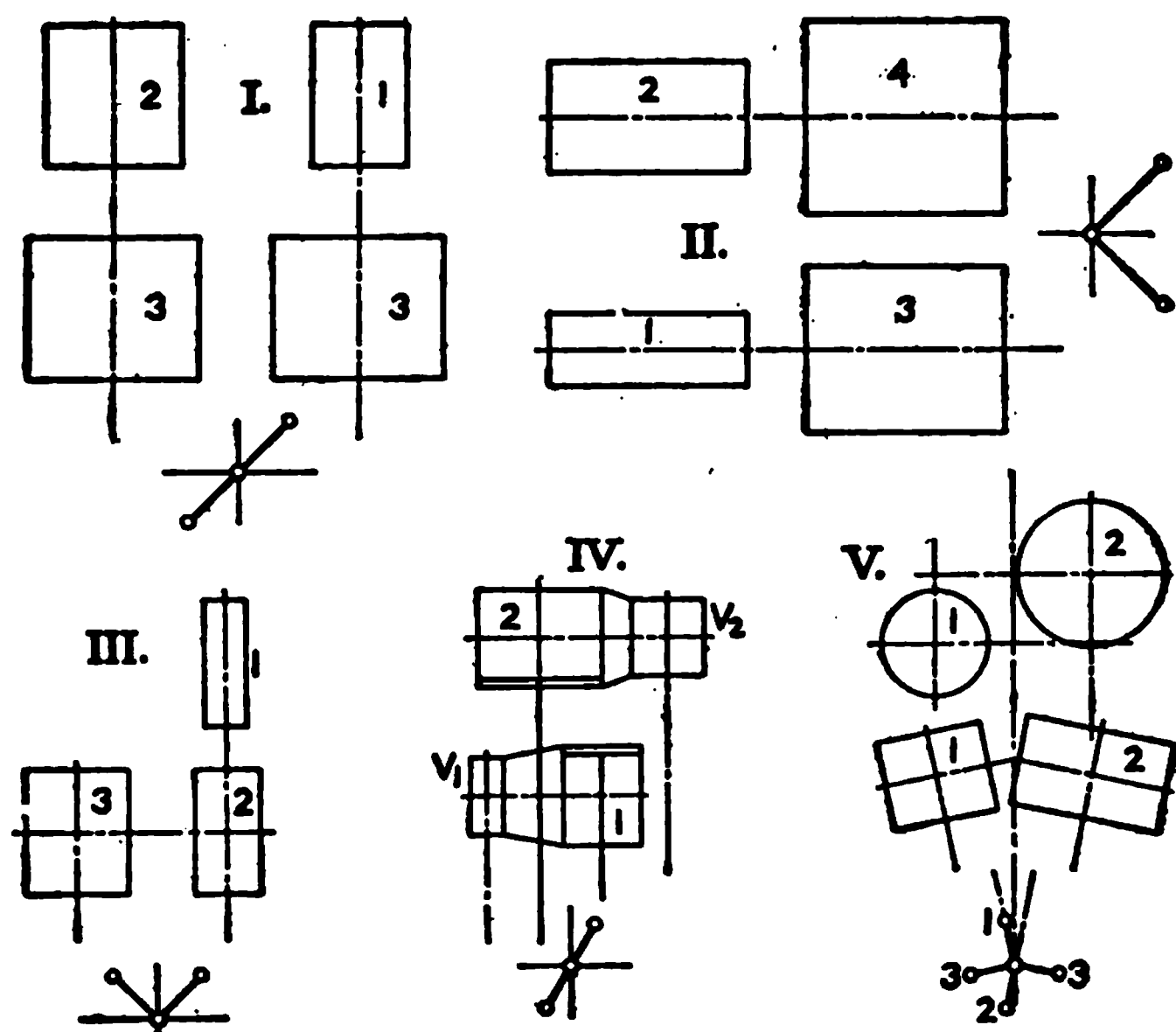


FIG. 613.—Special Arrangements.

- I. Triple Sulzer Generator-engine, Berlin, 1899; No. 12, Table 70 B.
- II. Quadruple Allis Engine, Chicago Exposition, 1893; No. 15.
- III. Two-crank Triple, Fraser & Chalmers, 1894; No. 13.
- IV. Collman Balanced Engine, Vienna, 1894; 22.1, 35.4×27.1.
- V. Torpedo Boat *Daring*, Thornycroft, 1894; 19, 27, 27-27×16.

An inspection of the sketches in Figs. 613 and 614 will show that the special arrangements in engines 33, 34, and 37 call for equal-work stages, while in Nos. 13 and 39 the first two stages should be smaller, since they are combined on one crank.

(d) VARIOUS CYLINDER-ARRANGEMENTS.—The obvious arrangement for an engine having more than two cylinders, especially if



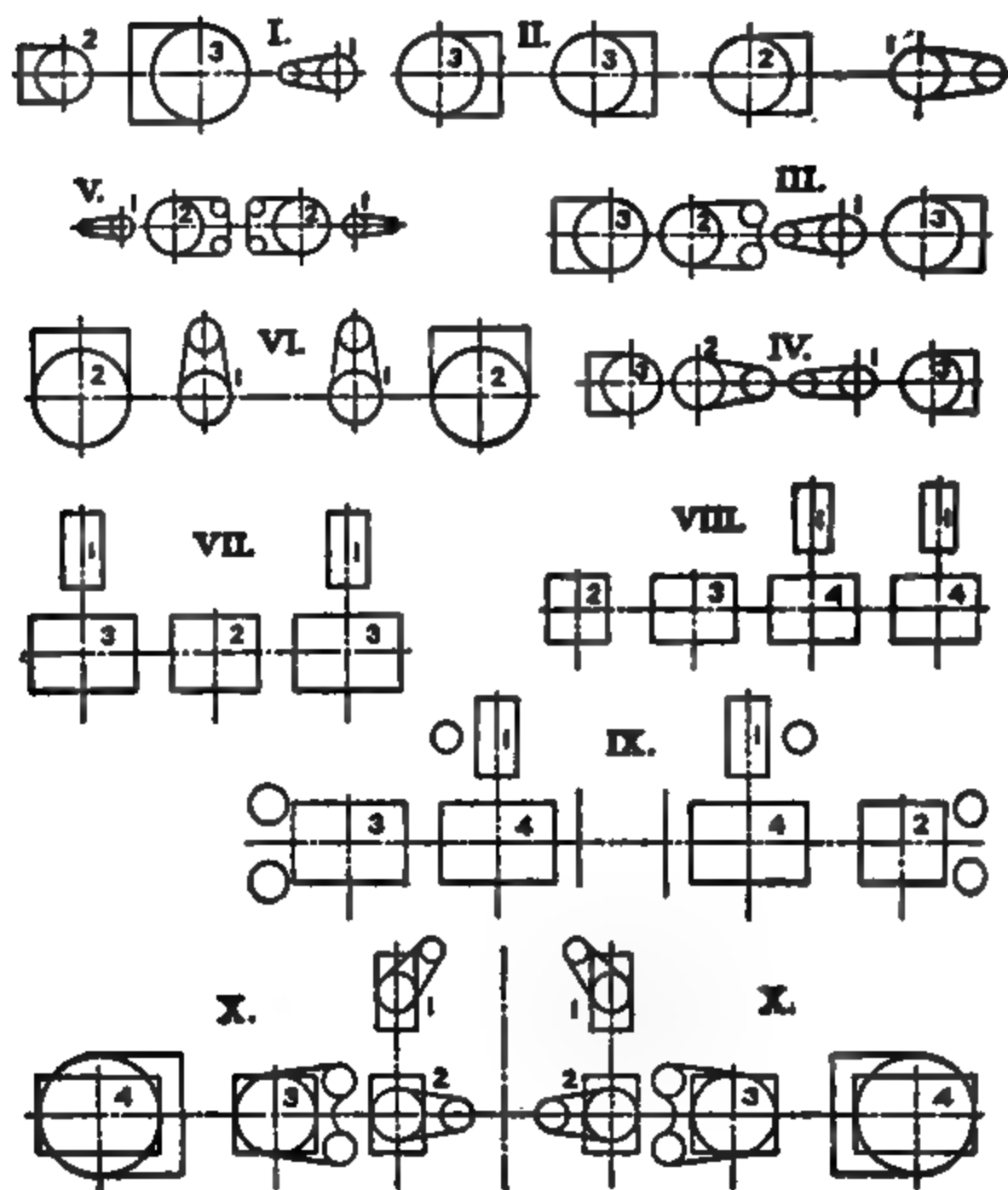


FIG. 614.—Marine Engines.

- I. Pacific S.S. *Minnesota*, 1902; No. 40, Table 70 C; 29, 51, 89×54.
- II. British Cruiser *Powerful*, 1896; 45, 70, 76-76×48.
- III. U. S. Battleship *Virginia*, 1904; 35, 57, 66-66×48.
- IV. British B. S. *Triumph*, 2d class, 1903, 29, 47, 54-54×39; see Fig. 241.
- V. Ferryboat, 1904, No. 10; duplex 22.5, 50×30.
- VI. Italian B. S. *Silvia*, 1895, No. 9; duplex 47, 89×51.
- VII. Steamers *Campania*, *Lucania*, 1892, No. 33.
- VIII. Steamers *St. Paul*, *St. Louis*, 1893, No. 24.



each contains a stage of the expansion, is to put them in a row in sequence, as in Figs. 229, 230, 234, and 259. For the sake of compactness, however, and to diminish the amount of mechanism required, the tandem is often combined with the side-by-side arrangement, as in Figs. 613 I., II., and III. (and in Fig. 606); I. and III. are typical schemes for a triple engine, the first being the more symmetrical. Fig. 613 II. represents a tendency which reached its height at about the time that engine was built; a little experience showed that there was no economic justification for quadruple expansion in stationary service. The last two diagrams in Fig. 613 show a scheme for making the engine self-balanced, by crowding close together the axes of two cylinders which work on cranks that are opposite (or effectively so).

Of the marine-engine arrangements in Fig. 614 I. shows an exception to sequence order in a three-crank engine, while II. is an example of plain sequence in a four-crank triple. Largely from considerations of balancing, the latter very common type of engine is now nearly always built in the form shown at III. and IV., with two closely-spaced groups and with the low-pressure cylinders outside.

One of the last large compound engines built for a sea-going ship is represented at VI., with a recent example of the same type for ferryboat service at V.; the former has the Joy valve-gear, which accounts for the unusual position of the valves. For paddle-wheels the plain quarter-crank compound is very common, with the cylinders at the bottom and the shaft at the top of an incline. When compound engines were standard practice for screw-engines they were often made with three cylinders and cranks, the H.P. between the two L.P.'s.

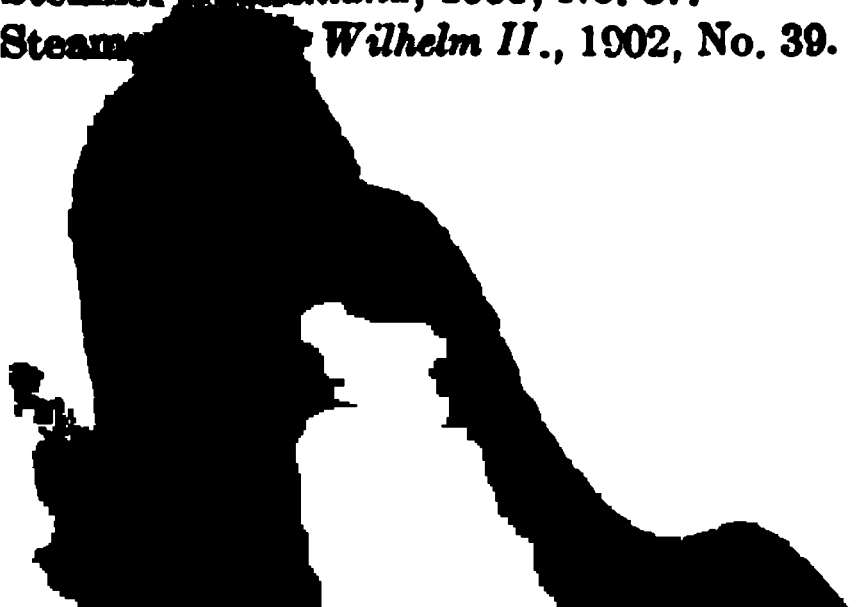
The last four sketches in Fig. 614 show rather special arrangements developed for large passenger steamers. Numbers VII., VIII., and IX. all embody the idea of dividing the last stage in order to avoid the use of excessively large cylinders, and then

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FIG. 614—*Title continued.*

IX. Steamer *Deutschland*, 1900, No. 37.

X. Steamer *Wilhelm II.*, 1902, No. 39.



dividing the first stage also, that the work-units may be equal. No. X. is like Fig. 613 III. in calling for smaller work-values of the first stages if the cranks are to receive equal powers.

All the diagrams in Fig. 614 are drawn to the same scale, and all but VII. and VIII. show the true spacing of cylinders and valves, thus giving an idea of the relative space needed by the various engines.

## CHAPTER XII.

### THE STEAM-TURBINE.

#### § 71. General Form and Action of the Turbine.

(a) STEAM-ACTION IN THE ENGINE AND IN THE TURBINE.—In the ordinary pressure-engine, the elastic force of steam is directly applied to the doing of useful work, through its action upon a moving surface in constrained expansion. When the confining reaction of the working element (the piston) thus balances the internal stress in the steam, we have what may properly be called a static-pressure cycle.

In the turbine, there is first the operation of transforming pressure-energy into kinetic energy of the jet, through free expansion—that is, expansion in which the internal pressure acts to accelerate the steam, instead of being held in equilibrium by a confining force. Then, the jet having been formed, its kinetic energy is applied to the work of driving the rotor, by means of dynamic pressure upon moving vanes.\* The first part of the process is predominantly thermodynamic, the second chiefly mechanical; and the whole constitutes a dynamic-force cycle.

The conditions which give maximum thermodynamic efficiency, for both types of machine, are set forth in §§ 14 to 16 and in § 24. The ideally perfect Carnot cycle having been modified by the omission of adiabatic compression up to the higher temperature, the resulting Rankine cycle (called Cycle B in § 16) is then the limit of effective performance, either in work upon the piston or in acceleration of the steam.

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\* In the reaction turbine these two operations are, in part, simultaneous; but that fact does not weaken the logic of putting them in sequence as parts of the energy-transformation process.

With the losses of effect in the cylinder, these losses being due to free expansion and interchanges within the cylinder. The steam is wasted in moving the steam, or thrown away above the exhaust-pressure; the second law of condensation, which is, of course, the most widely from ideal perfection.

The simplest theory of the turbine requires that the operation of the turbine be adiabatic and frictionless and that in operation no work shall be lost in friction. At first to ignore secondary actions leads to a misunderstanding of the function of the turbine and thereafter it is comparatively easy to add which shall represent the influences of secondary special experiment.

In the actual turbine, harmful secondary thermal nature are relatively insignificant. The gain over the engine must be placed in the balance and in wasted movements of the steam. The loss of these is to change potentially effective heat at lower temperature, in which steam shall be rejected in the exhaust. On the whole, the total steam-cycle losses in turbine are in the same class as to magnitude, with the advantage for the turbine in the possibility of high efficiency with less resort to special experiment.

Numerical data in regard to this matter are given in XIII.

(b) CLASSIFICATION OF STEAM-TURBINES.—The classification is chiefly upon differences in the manner of operation, steam-velocity, these differences being the resulting variations in form and function.

*Operation in Stages.*—A matter of importance is the speed of the rotor of the turbine, both in operation and safely the very high speed of the steam issues from the nozzle is always a

of meeting this difficulty is to diminish the steam-velocity, by dividing the expansion or pressure-drop into a number of steps or stages, quite closely analogous to those in a multiple-expansion engine, but likely to be much more numerous. The other scheme is to divide the energy-absorption process into several stages, by causing the fully formed jet to impinge upon two or more sets of moving vanes in succession. The terms "pressure-stage" and "velocity-stage" are rather awkward for continual use: to get shorter words we shall let "stage" stand for pressure-stage, and speak of one or more "impulses" instead of velocity-stages. The latter usage is further recommended by the fact that only in what are called impulse turbines can the energy from a given nozzle (or group of nozzles) be absorbed in portions.\*

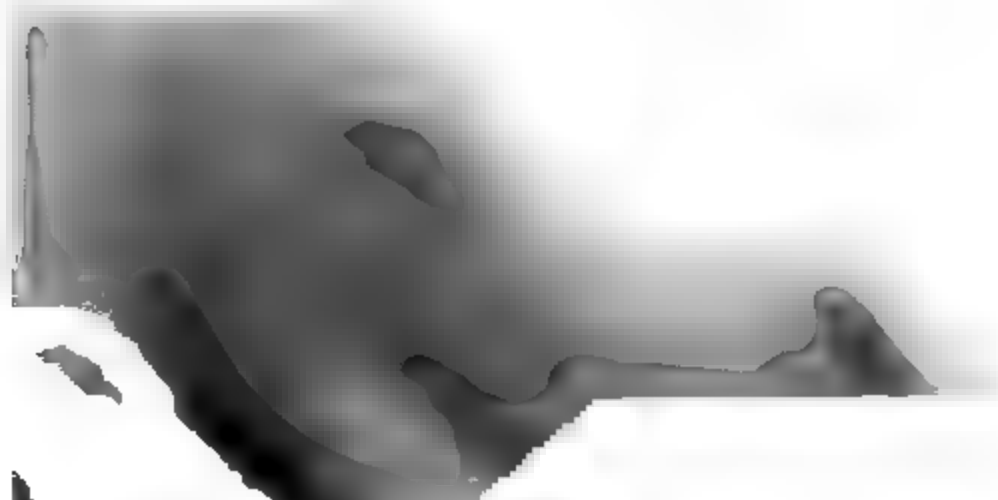
*Impulse and Reaction Turbines.*—Another important distinction depends upon the question whether velocity-generation takes place wholly within the fixed nozzles, or whether it is also combined with the function of absorption in the moving element. The impulse turbine, just mentioned, has these two functions kept separate: then between any two successive distributors or sets of nozzles the steam is under uniform pressure, and the current has practically the same relative linear velocity at entering and at leaving any set of moving vanes. In the reaction turbine, on the other hand, pressure-drop and acceleration take place in the moving element as well as in the fixed element, so that the steam leaves the vanes with a higher relative velocity than it had at entrance. The terms "impulse" and "reaction" will be more closely defined in § 72, with some discussion as to how near they come to being truly descriptive when thus applied to steam turbines.

In the matter of general steam-action, then, we have the major classes of impulse and reaction turbines; and in the first class we must further distinguish between the cases of one or of more than one impulse or velocity-stage within each pressure-stage.

*General Form and Arrangement.*—The primary purpose of the following examples (Figs. 616 to 631) is to give a clear idea of

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\* It will be noted that the terminology "compound" and "complex", tentatively put forth in § 29 (f), is here definitely abandoned.



the essential form of the elements that It is most usual to make the steam direction of flow lie mainly parallel only example of a radial-flow turbine Fig. 627. With axial flow, the working guides, vanes or buckets) lie within tion" (in the geometrical sense): and 623, and 628 are "developed" or unfolded through these working-elements.

A marked distinction as to form cellular construction of the multiple 620, 621, etc., and the drum arrangement Fig. 629. And besides a progressive the rotor, most marked in Figs. 629 a sometimes divided into two sections—sure—in separate casings or "cylinders" these divisions (of the rotor) are on the practice the H.P. and L.P. turbines are propellers.

The machines described in this section with regard to the essential character



FIG. 616. - Element of Single-stage One-imp

and not according to their commercial engineering importance, and no attempt order in development.

(c) THE SINGLE-STAGE ONE-IMPULSE turbine of this type that has been constructed De Laval, of which a brief partial description The essential form of the working-elements while Fig. 617 is the section of a complete are used which have to be run at the



order to get the peripheral velocity necessary to reasonably good efficiency—practice ranging from a 4-inch diameter at 30,000 R.P.M. for 5 horse-power to 30 ins. at 11,000 R.P.M. for 300 horse-power, these diameters being measured to mid-length of the vanes. Then gearing is used to reduce the speed to something practically applicable at the power-shaft, G on Fig. 617. Other

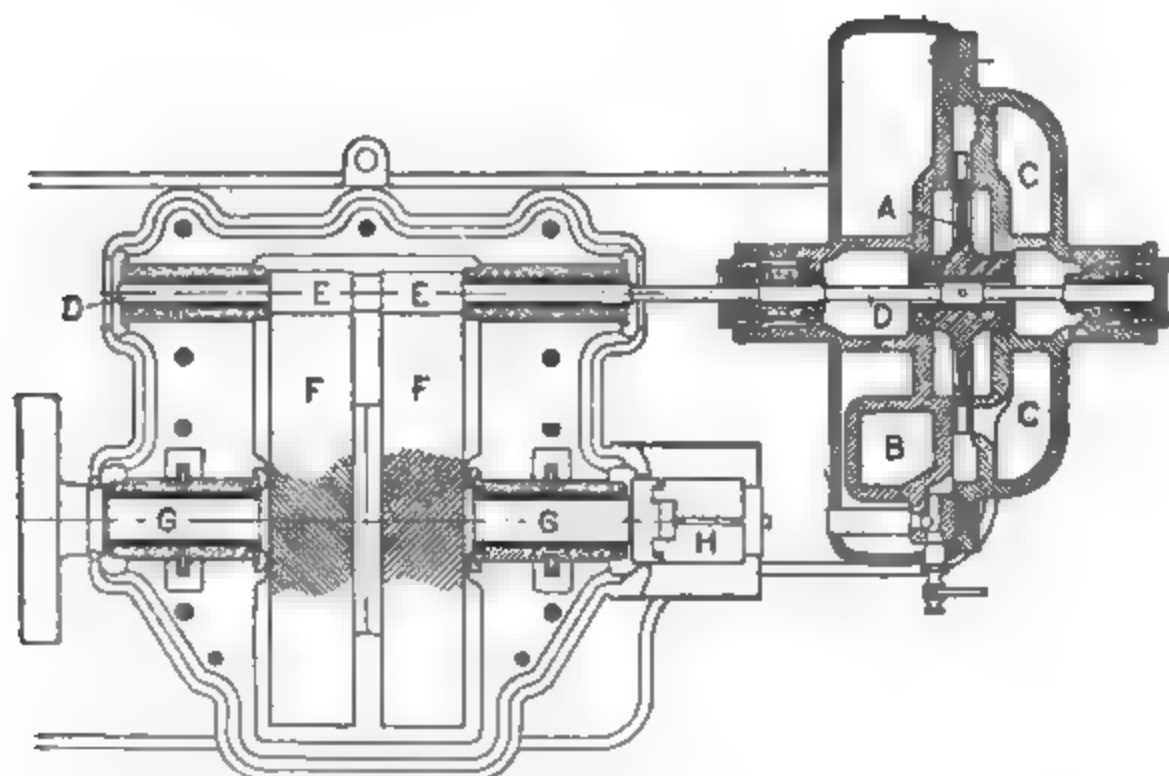


FIG. 617.—Section of 30 Horse-power De Laval Turbine, with wheel about 8" in diameter at 20,000 R.P.M., vane-velocity about 700 ft. per sec.

single-stage turbines have been built for only moderately high rotary speed, with much larger wheels, one example being described in (h), under Fig. 625: but the designers of these have all gone on to the multiplication or combination of pressure-stages and velocity-stages.

The turbine-wheel A, Fig. 617, is a solid steel disk of special form, designed so as best to resist centrifugal force. Since it is impossible to make a wheel with absolutely perfect balance, or with the center of mass exactly on the geometric axis, the wheel is mounted on a light flexible shaft D. This shaft can easily deflect enough to let the center of mass come to the axis of rotation, whereupon the wheel will spin without vibration. Of course, the

mechanical balance is made as correct as is practically possible, the flexible shaft is expected to care for only a small residual eccentricity.

The steam-chamber which supplies the nozzles is shown at B, the exhaust-chamber at C. The smallest turbines have only one nozzle; but as diameter and power increase more nozzles are used, the 30-in. wheel having perhaps twelve. These are usually spaced equally about the circumference, chamber B being given an annular form; but in the latest practice the nozzles are sometimes arranged in close groups, to diminish eddy-current losses. A throttling governor controls the admission of steam to the turbine.

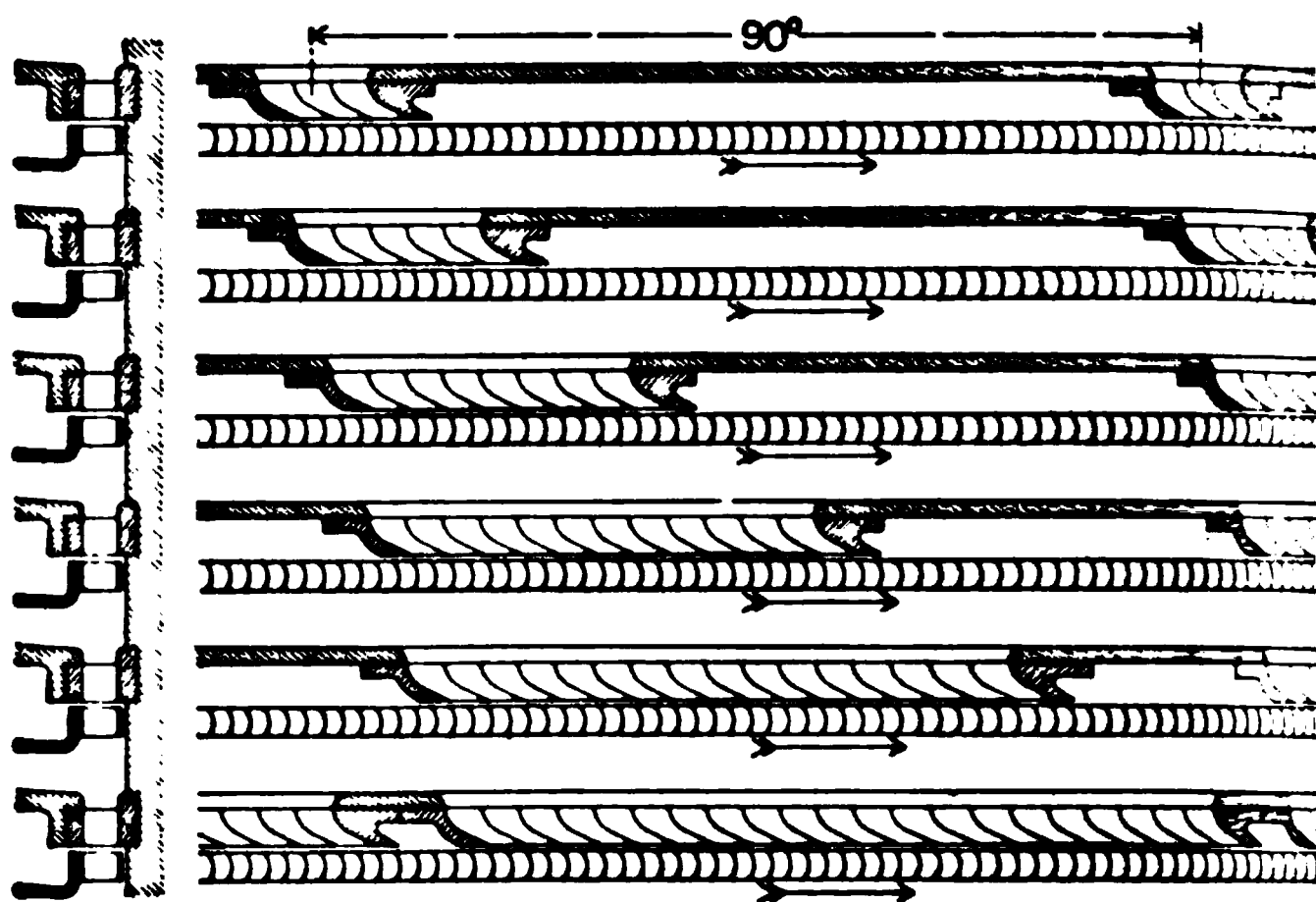


FIG. 618.—Developed Section of Multiple-stage Impulse Turbine with partial peripheral admission: based on high-pressure end of Rateau design Fig. 620, but with rate of expansion somewhat exaggerated.

but when there are a number of nozzles, all but one or two are provided with valves, having external handles; then the power of the turbine can be adjusted by hand to suit the load, only a small range of control being ordinarily left to the governor. For a light load, it is far more economical to have a few nozzles working with high pressure than to have all open and the steam greatly throttled by the governor.



Sometimes a De Laval turbine which is to run either condensing or non-condensing will have two sets of nozzles, arranged alternately, either set to be turned on for the condition to which it is fitted. To develop the same power, a non-condensing nozzle must have a larger throat and less expansion or flare than one for vacuum-exhaust.

(d) **MULTIPLE-STAGE ONE-IMPULSE TURBINES.**—Representative forms of the working-elements of turbines of this class are diagrammed in Figs. 618 and 619; while a typical longitudinal section is given

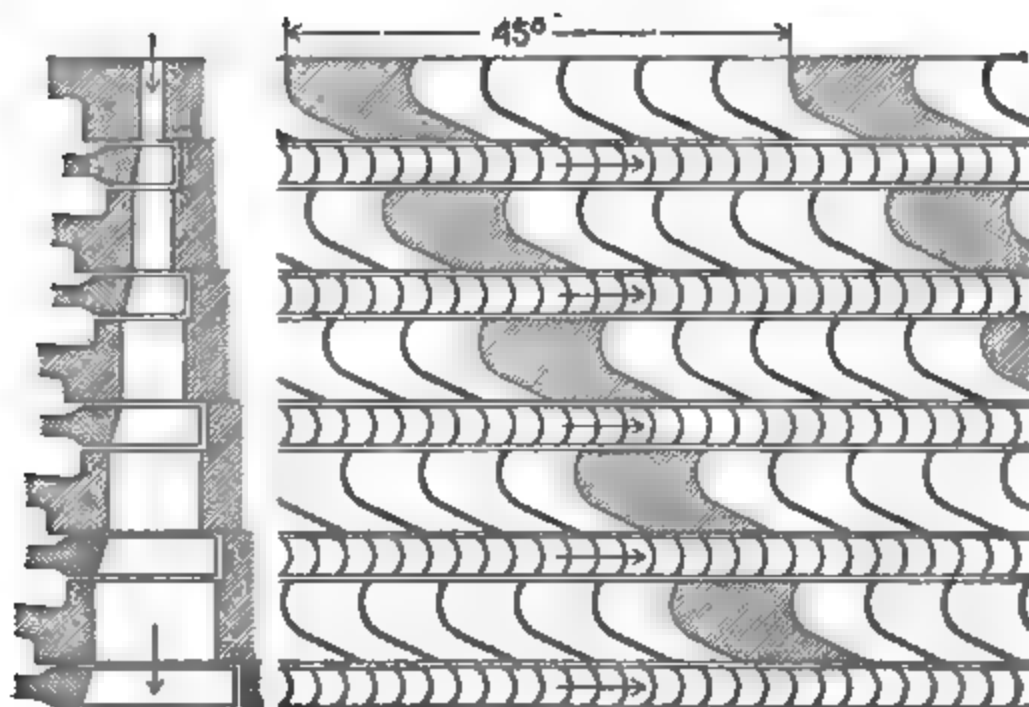


FIG. 619.—Developed Sectional View of Multiple-stage Impulse Turbine with full peripheral admission: cutline of the low-pressure division of a Zoelly turbine.

in Fig. 620. With any considerable number of stages, the pressure-drop in each is so small that there is no need for the diverging nozzle; then the distributor takes the form here indicated, with rectangular openings divided by curved guide-plates into what are, in effect, groups of converging nozzles. The first two figures illustrate an important distinction, by showing the two methods available for increasing the sectional area of the passage for steam as the expansion progresses. In Fig. 618, partial peripheral admission is used, and, with a constant radial depth of the vanes and guides, increase of area is secured by making the successive groups

wider in the circumferential direction. The other method is to use the whole circumference for the passage of steam, and to increase the radial depth as necessary.

A fact made evident by these drawings is that in the so-called axial-flow turbines the steam-current really follows a slightly helical path. The method of plotting this path and of determining the angular advance of the successive distributors will be found in § 7. (k).

Another distinction to be made between Figs. 618 and 619 is based on a difference in the manner of passing the steam from one element to the next. In the first case, the steam from any wheel is discharged into a sort of receiver-space, in front of the next distributor; and the current is thus given a good chance to settle and eddies in the (comparatively speaking) stationary mass of steam that fills all the space about the wheel. But with close and direct discharge into the next distributor, as in Fig. 619 it appears

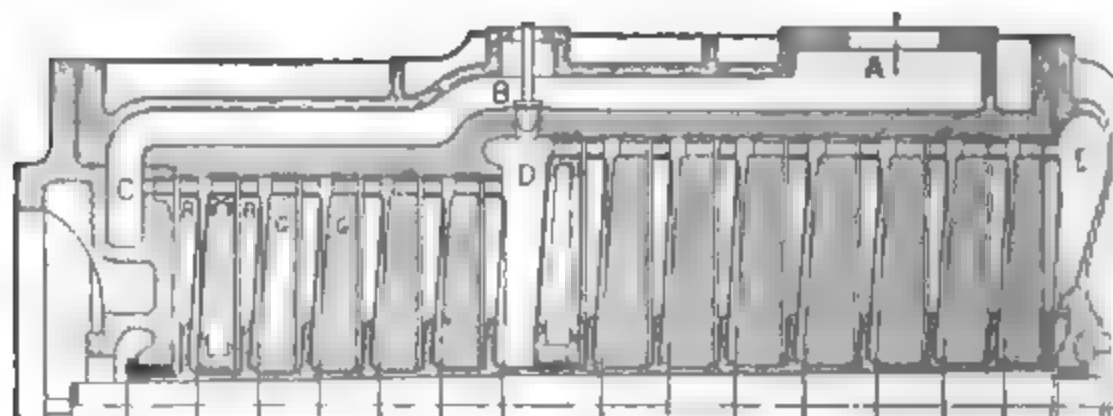


FIG. 620.—Section of a 24-stage Rateau Turbine, in three steps and 12 cylinders: 500 H.P. at 2400 R.P.M.: mean diameter of vanes 20" to 33": vane-velocity, from 220 to 345 ft. per sec.

that this possibility is reduced to a minimum, so that there is no reason to expect the fullest attainable utilization of the residual velocity from one stage in helping to produce effective initial velocity in the next stage.

It may be well to remark that Fig. 618 does not show the structural form, but that Fig. 619 is a correct drawing, so far as it goes into detail. The necessity of sufficient strength of metal to carry the diaphragm (subjected to different pressures on its two

sides) accounts for the stout cross-struts between the groups of nozzles in Fig. 619.

(e) THE RATEAU TURBINE may well be taken as typical of the many-stage single-impulse class. The two-cylinder turbine partly shown by Fig. 620 has rather a high number of stages: a later design illustrated by Stodola, for 500 H.P. at 3000 R.P.M., has 13 stages in one cylinder, with wheels approximately 18" to 30" in diameter, and velocity ranging from 250 to 400 ft. per second.

In Fig. 620 steam enters from the automatic throttle-valve at A and passes to the first nozzles at C: these form one group, subtending only a small angle; and the constant vane-length shows that partial peripheral admission is used all through the first two groups of wheels, or the first two diameter-steps, to E. At B is a hand-controlled bypass valve, which can be opened when the turbine must carry a heavy overload. Between the cylinder-heads

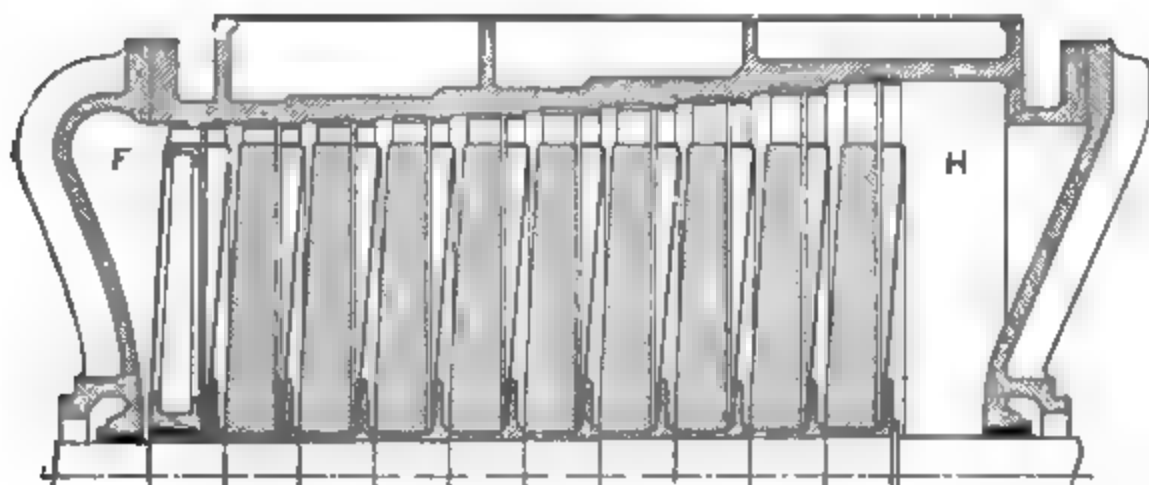


FIG. 620—Continued.

E and F is space for a double bearing and the governor, steam passing from E to F through a pipe below the floor-level. In the last step of ten stages the nozzles cover the whole circumference (after the first few stages, at least), increase in area of passage being secured by making the radial depth greater. From H there is a short and direct connection to the condenser.

As to the construction of this turbine, it will be noted that each wheel is a disk of steel plate, riveted to a light hub and flanged at the outer edge so as to form a narrow cylindrical rim upon which the vanes are riveted: in the larger diameters, an extra disk of

plate is usually added, for the sake of great stiffness. Holes through the wheels insure equalization of pressure on the two sides. The partition walls, in form fixed wheels held by the casing are shown in detail only at the beginning of each step: these walls are made up of light cast frames or spiders, covered with discs of plate. Leakage from one cell to the next could take place only at the shaft, where it is prevented by the close fit of the bronze bushings—labyrinth packing (see § 74 (f)) being also used between the higher stages in the later designs.

In this, as in most horizontal turbines with many stages, the casing is divided along a horizontal plane through the axis, and the top half can be lifted off, carrying with it the upper half of each diaphragm. It is evident that careful fitting will be needed to insure tightness where the two parts of these diaphragms meet.

In none of the drawings in this section is much attention paid to non-essential mechanical details, such as bearings, stuffing-boxes, valves, etc. A few good examples of these parts will be found in § 74.

(f) **THE MULTIPLE-IMPULSE TURBINE.**—The idea of abstracting velocity from a jet of steam by making it impinge upon several sets of vanes in succession offers a very attractive means of reducing the velocity at which the vanes themselves must move—in simple theory, more effective than an equal multiplicity of pressure-stages, as is brought out in § 72 (f): but it is also made clear there that if the velocity-stages be too numerous, secondary losses due to friction and eddies will become excessive. In practice, four impulses from one initial velocity is the highest number that has been used in a successful turbine; and as the result of fuller experience there is evident a strong tendency to cut this down to two.

The three examples which follow not only illustrate the application of this particular principle, but are also of general interest as representing three contrasting types in the form and arrangement of the turbine-element. The Curtis turbine has curved radial vanes, with side admission, as in the examples already given and in the reaction turbine also; the Riedler-Stumpf shows the Pelton-wheel type of bucket, with tangential admission; while the Elektra is a case of radial flow, with vanes parallel to the axis and repeated

impulse upon the same set of vanes. Detailed discussion of the action of the vanes and buckets will be found in § 72 (c) to (f).

(g) THE CURTIS TURBINE is the most prominent member of the multiple-impulse class. In the smaller sizes, below 500 Kw.

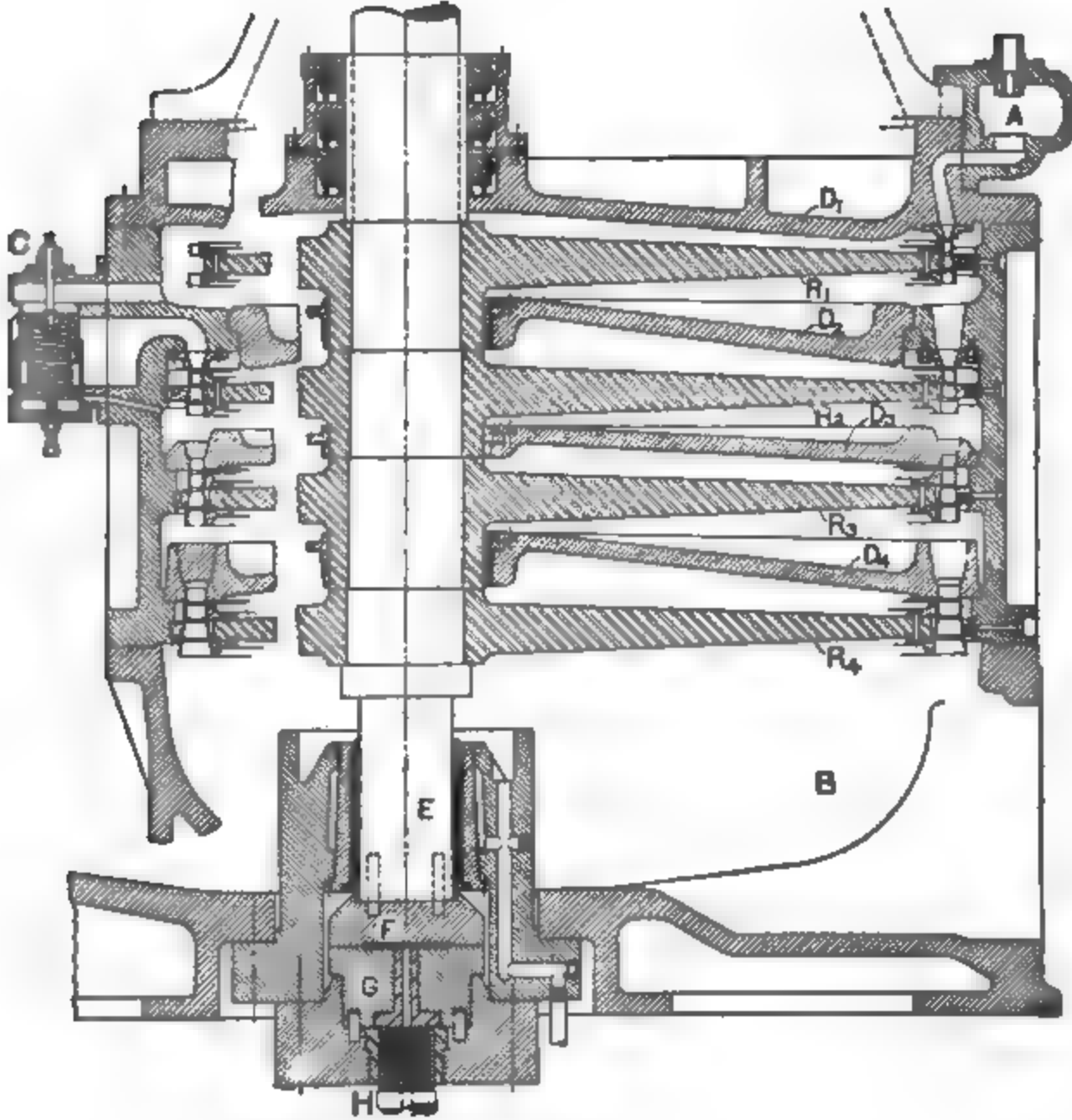


FIG. 621.—Section of a 2000 Kw. Curtis Turbine, with four two-impulse stages; 100" wheels at 750 R.P.M., vane-speed about 325 ft. per sec.

capacity, this machine is made with the axis horizontal; but the larger units, running up to 8000 Kw., are always vertical, with the generator, of course, at the top. A good typical recent design is shown in section by Fig. 621. which gives a very fair idea of

the larger detail in the construction and the wheels are heavy and the blade-rings, made in a number of segments, cast-iron diaphragms are undivided, so must be assembled at the same time. a part of the cylindrical casting, which is removed, can be removed.

The whole weight of the revolving generator-field, is carried by the large has its working faces on the hard cast-iron the hole at the axis of the big set-screw the height of G, oil or water is pumped recess between the blocks; and as it is a thin film, separating the metal surfaces lubrication; in fact, the revolving mass is liquid. When oil is used, the casing is closed at the top, with suitable packing getting into the exhaust-steam: here is water-lubrication, and the water is supplied the exhaust-base, the shells of the side to afford a free passage and prevent To the spindle H is keyed a worm-wheel bearing can easily be adjusted from the

The admission of steam is controlled at A, each supplying one or two nozzle valves are opened, through the action demand for power varies; and the turbine simple throttling, but by something motion of cut-off in an engine. Details of in § 74, at Fig. 678.

Quite often there are two groups of ends of a diameter: but here there is circular dimensions of the successive circles indicated by the diagram-sketch in Fig. in the second ring, marked N<sub>B</sub>, is served valve at C in Fig. 621, shown in detail in that the disk of this valve feels the pressure

top face, that in chamber 2 on the bottom; and when the difference between these exceeds a certain amount the valve will open. Since, however, this is below the main admission, it will affect only the distribution of work and not the total power; but it does come into service when the load is heavy and more nozzles are open in ring  $N_1$  than can properly be taken care of by the normal part of  $N_2$ .

It will be noted that the partial sectional view at the left of Fig. 621 is not a continuation of the main section, but is made by a plane about at right angles to that of the latter view.

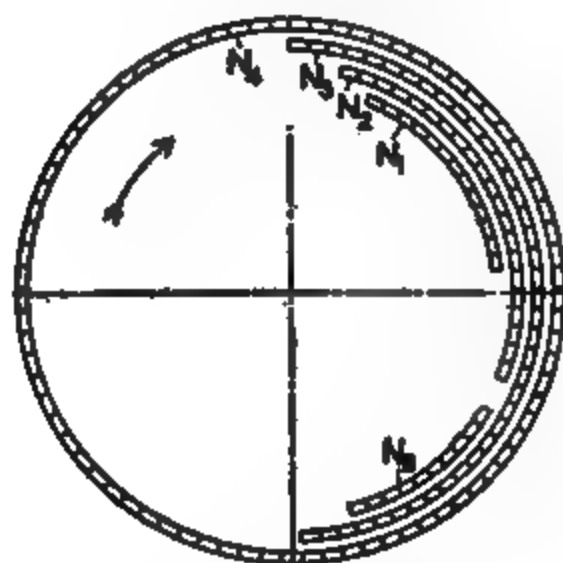


FIG. 622.—Diagram of the Nozzle-admissions in the Curtis Turbine.

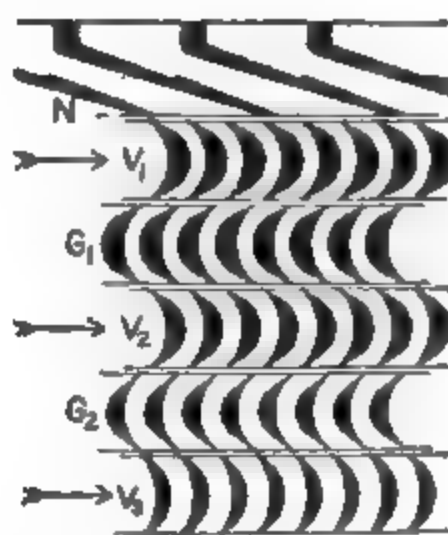


FIG. 623.—Three-impulse Element of a Curtis Turbine.

To give an idea of the shape of the vanes, the section of a three-impulse element is given at Fig. 623, reproduced from Fig. 73. Smaller Curtis turbines are made with only one or two stages, with three or four impulses in each. Some of the earlier very large machines had two four-impulse stages. Now, however, the type of design in Fig. 621 is becoming the standard for all big units.

(h) THE TANGENTIAL-ADMISSION WHEEL, similar to the Pelton water-wheel, has been used by several inventors of steam-turbines, and most extensively developed in the Riedler-Stump design, of which a number of different working-elements are sketched in Fig. 624. At I. we see how the flat semicircular buckets are milled in the rim of the steel wheel, each at an angle of  $15^\circ$  or  $16^\circ$  with a tangent to the circumference at its mouth. Wheels are made with

either single or double bucket-rows, as shown at II., which also gives the arrangement of the nozzles—these nozzles being circular at the throat, but changing to a rectangular cross-section at the outlet.

Various devices for securing two impulses in a stage are shown at III., IV., and V. In each case the curved guide G receives the

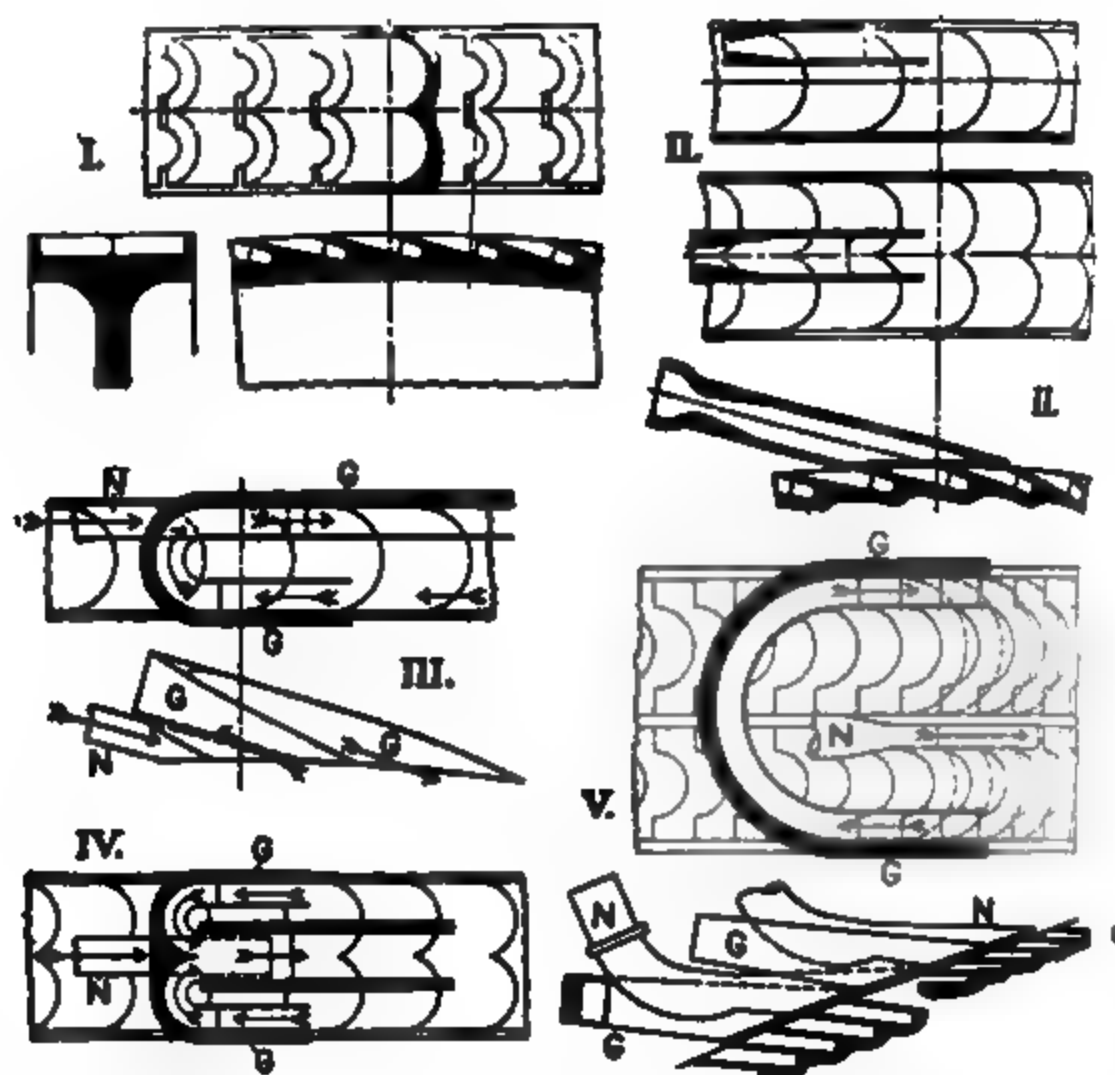


FIG. 624.—Working-elements of the Riedler-Stumpf Turbine.

steam as it escapes from the buckets after the first reversal of velocity, and turns the current back for a second entrance, either into the same buckets, as in III. or IV., or into another larger set as in V. With return to the same buckets, the guide must have the peculiar helical shape shown by the side view at III., because the absolute steam-velocities at entrance and at exit will make different angles with the wheel-rim—see Fig. 640: but at V. the



second-stage buckets are given more slant, so that the guides can be flat. These guides are built up of pieces of plate, stamped out and riveted together, the heavy outer walls which give stiffness to the whole and which are fastened to the carrying brackets being usually made up of several layers of thick plate: this laminated construction is the more necessary when they must be bent to a special form as at III. or IV.

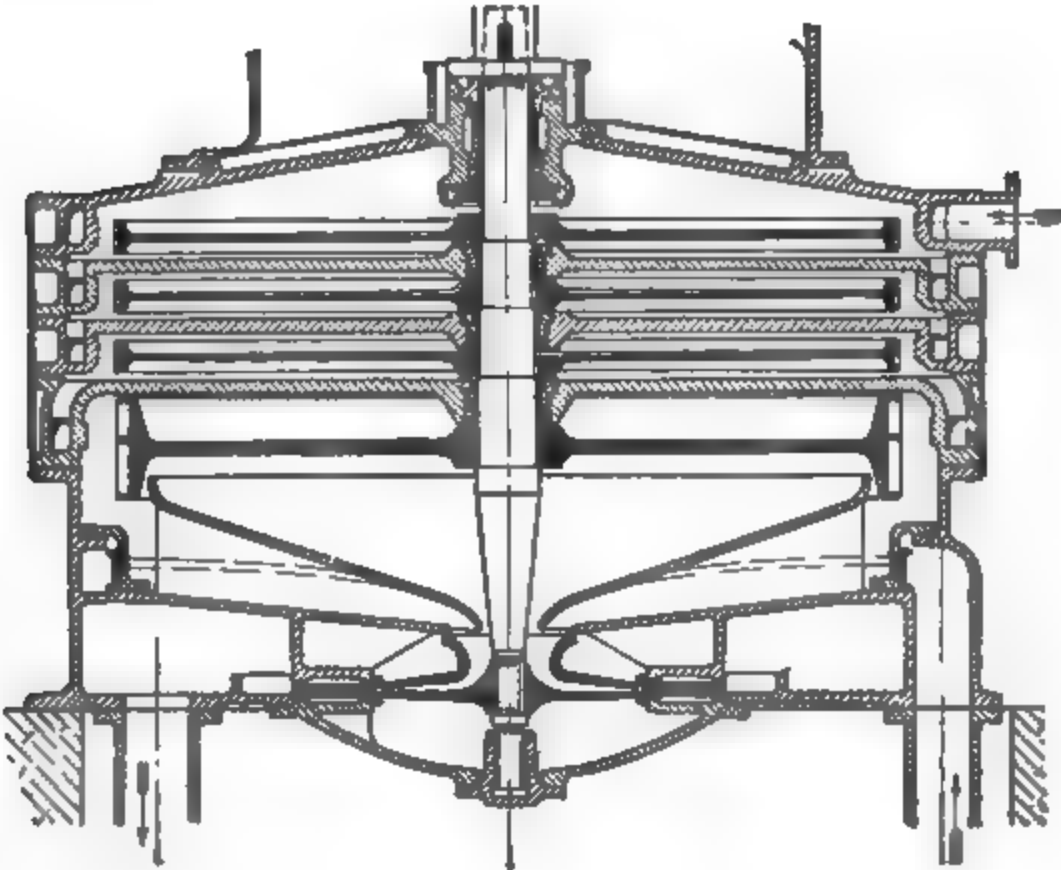


FIG. 625.—Vertical Four-stage Two-impulse Riedler-Stumpf Turbine.

The Riedler-Stumpf turbine is made in a number of different forms, ranging from a small one-stage horizontal machine to the large vertical design in Fig. 625, which is very much like the Curtis in Fig. 621 as to its general arrangement. Interesting particulars of a single-stage turbine built as one of the earlier experimental machines have been published: the wheel was 78.8" in diameter and ran at 3000 R.P.M., so that the peripheral velocity was about 1030 ft. per sec.; there were between 80 and 90 nozzles, distributed all around the wheel; and the machine developed 2000 H.P., with very fair steam-economy. By multiplying stages and using the

two-impulse arrangement the rim-speed can be greatly reduced, and at the same time the efficiency of the apparatus increased. The principle of partial peripheral admission can very easily be applied by varying the number of nozzles in any stage: to regulate the turbine, steam is admitted to more or fewer nozzles in the first stage.

An interesting feature of Fig. 625 is the centrifugal condenser-pump at the base, the jet-condenser being formed right at the base. Of course this device cannot maintain the high vacuum that is attainable with a surface condenser and good reciprocating pumps; but it is exceedingly simple, and has been credited with an absolute pressure of 0.1 atmosphere, equivalent to 27 inches vacuum.

Another type of tangential-wheel turbine is represented by the skeleton-diagram in Fig. 626, which closely resembles the Pelton wheel in having the impulse-cup buckets formed in separate castings, which are bolted to the rim of the wheel—this sketch indicates how the turbine section for steam flow can be increased by using two or three wheels side by side in the stages.

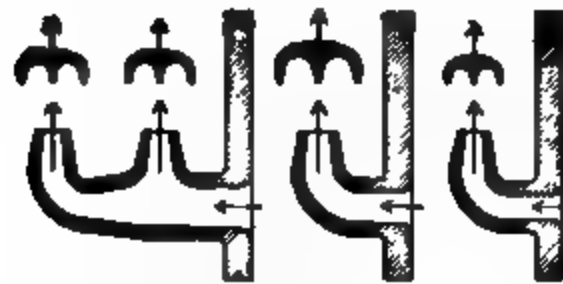


FIG. 626.—Elementary Sketch of the Kerr Turbine.

(1) A RADIAL-FLOW TURBINE with multiple impulse is shown in Fig. 627, where there is the further characteristic (to be emphasized in the present connection) that one vane-ring serves for the whole operation in the stage. The partial longitudinal section at B shows how the vanes project from the side of the wheel into an annular groove in the face of the diaphragm. On the high-pressure side, marked H, there are two nozzles, diametrically opposite, on the low-pressure side there are four. The manner of securing the three impulses is sufficiently obvious; and the general point of the general scheme is suggested by the question whether the steam-current will docilely follow the very intricate path set out for it. In the earliest patent of this arrangement (British, 1848—see Neilson's *The Steam Turbine*, 1903, page 2)

the steam-current is shown as acting upon the single ring of vanes twelve times, between inlet-nozzle and exhaust. It need hardly be remarked that experience would have entirely failed to justify this very complicated arrangement.

(j) THE PARSONS TURBINE.—The name Parsons is as closely associated with the multiple-stage reaction turbine as is the name

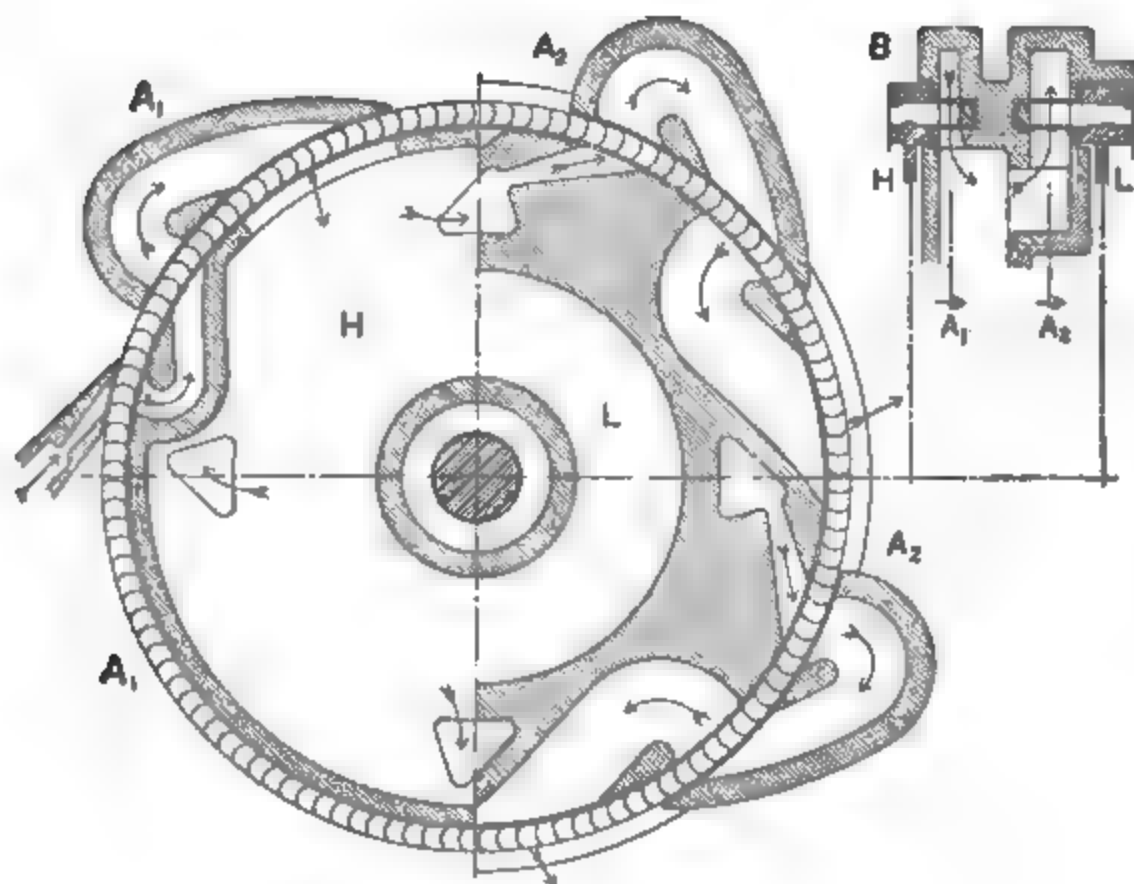


FIG. 627.—The Elektra Turbine, radial-flow, two-stage, four-impulse: much structural detail omitted.

Corliss with a type of engine—an association which will likewise probably persist long after patent-rights have expired. Partly because of the early pre-emption of the field and quite a long period of evolution to the arrangement in Fig. 629, partly because choice as to the form of the working-element is practically limited to the straight vane with side admission, there is really only this one reaction turbine in effective existence, as compared with a number of impulse types.

The name "reaction" comes from the fact that the major part

of the driving-force upon the rotor is due to the reaction of the current of steam which receives linear acceleration within the vanes



FIG. 628.—Element of the Parsons Turbine.

while only a small component is due to the velocity with which jets from the fixed vanes impinge upon the moving vanes—a full explanation of this action being given in § 72 (f). To fit this process the vanes have the shape shown

Fig. 628, where the steam flows from left to right, and the vanes have a much larger angle (from the line of movement) at entrance than at exit, and the channels between them narrow toward exit, thus acting as contracting nozzles. Further, since there are different pressures on the two sides of a ring of vanes on the rotor so that steam will flow through the whole ring, the fixed ring must likewise be complete, without any blank portions; in other words, partial peripheral admission is not practicable in a reaction turbine. It is for this reason that there is so much greater difference in size between the highest and lowest stages in Fig. 629 than in, for instance, Fig. 621—diameter of rotor as well as dimensions of the vanes entering into this question of size. The progressive increase in rotor-diameter is one of the prominent characteristics of this type of turbine.

In Fig. 629 the main admission-valve (automatic-throttling) is at  $V_1$ , and the steam normally begins its work at  $A_1$ ; but if the load rises above what can be carried with full pressure at  $A_1$ , the governor opens the bypass valve  $V_2$ , thereby admitting steam to the second step of the rotor,  $R_2$ . This cuts the first, small-diameter step more or less completely out of action, the blades simply churning a mass of steam which has little progressive motion; but the larger area of cross-section available at  $A_2$  lets in so much steam that increased power is developed on the two latter stages alone, though with lower thermodynamic efficiency of the turbine.\*

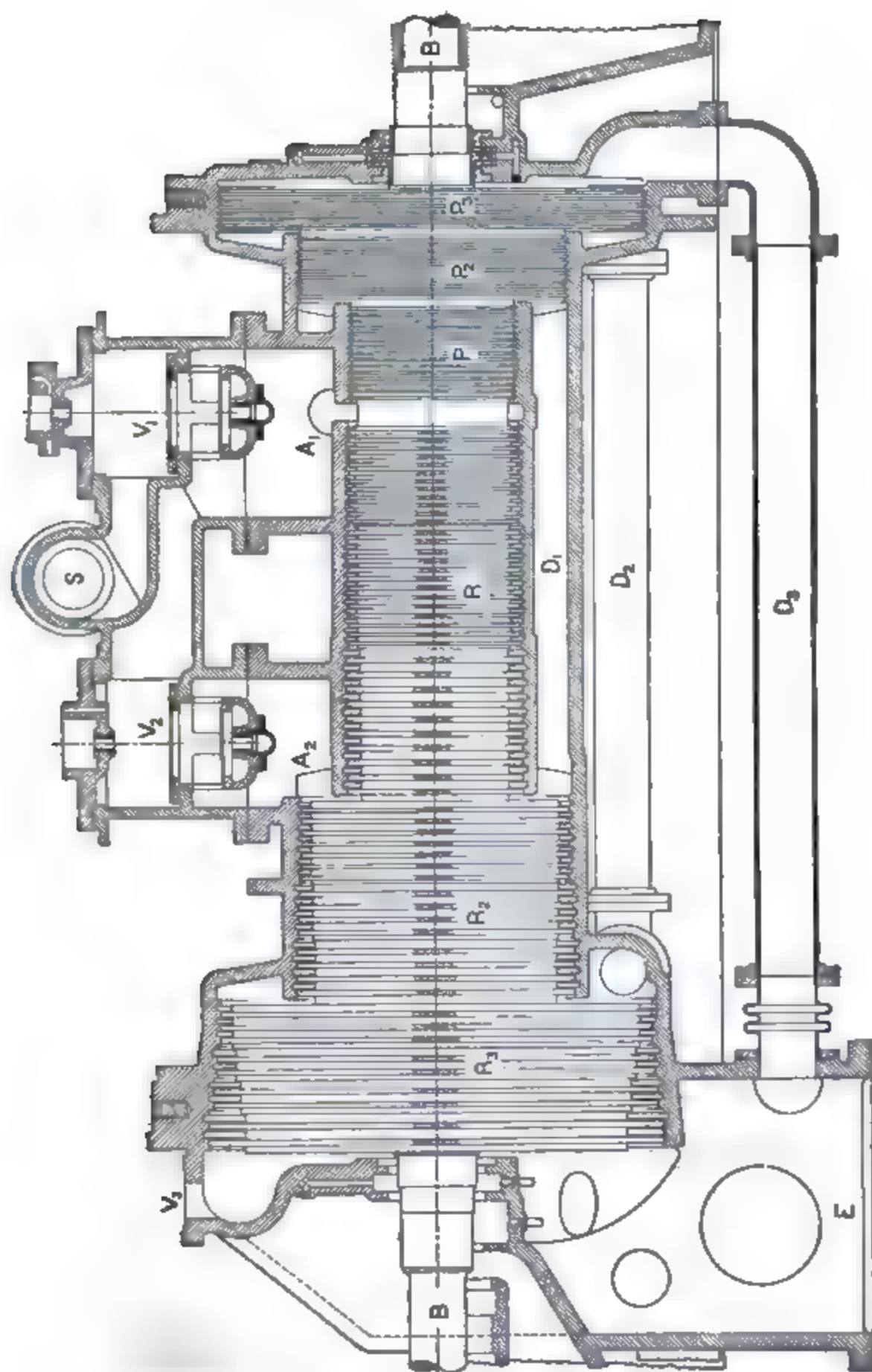


FIG. 629.—Section of

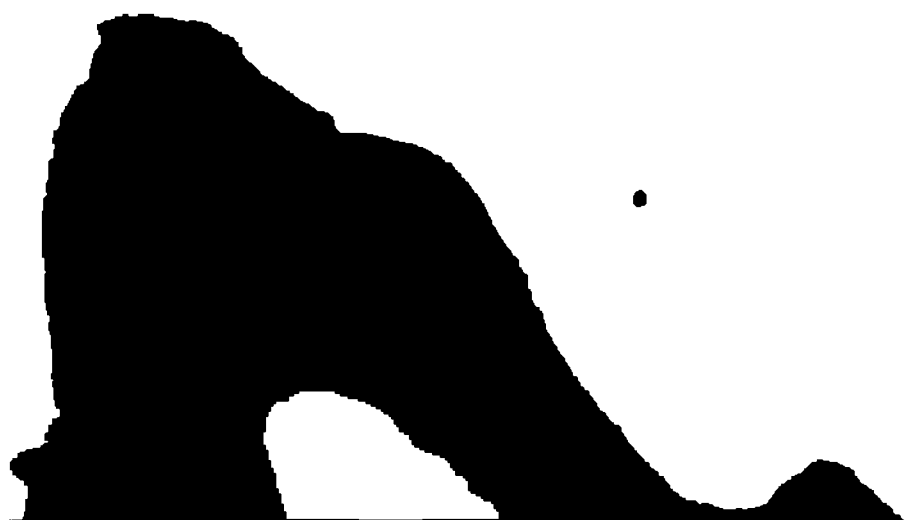
Westinghouse-Parsons Turbine, with 50 three steps.



Since there is a drop of pressure through each set of vanes, it is evident that there will be tendency for the steam to leak past their ends; consequently a close fit, or a very small end-clearance, is necessary. This requirement is practically one of the most severe that have to be met on the side of construction, for not only must there be careful original work, but there is also the danger that rotor or casing will be distorted by unequal heating or other cause so as to make the ends scrape, or even to break the blades. To compensate for this disadvantage in comparison with the impulse turbine, which has so very much smaller possible openings for leakage, appears the advantage that the reaction turbine does not have a great deal of wheel-surface exposed to the frictional drag of surrounding steam.

(k) **TURBINES OF MIXED TYPE.**—The latest development in turbine design is the combination of high-pressure stages working by impulse with low-pressure stages of the reaction type. Two good examples are given in Figs. 630 and 631, the first having two impulse stages just like those of the Curtis turbine, the second having tangential buckets as in Fig. 624, while the low-pressure stages in both are of the Parsons type. The Sulzer turbine is so arranged and proportioned that it is self-balanced as to endwise steam-pressure, thus obviating the troublesome balance-piston: in the Union design the upward steam-thrust acts against the weight of the rotor and serves the very useful purpose of relieving the step-bearing of pressure—in fact, provision is made at one bearing for holding down the rotor if necessary.

(l) **REVERSIBLE TURBINES.**—The feature which most distinguishes marine turbines from those for stationary service is the provision that must be made for running them backward. Vanes properly formed for forward motion cannot be used for reversed driving. The most obvious scheme is to combine with the main turbine a smaller turbine for backward running. This is not necessarily of less power, but is made compact and with but a few stages, hence of comparatively low efficiency. It can be added to the low-pressure end of the main rotor, where the frictional resistance to the normally idle (useless) movement of these extra parts will be very small.



The division of the turbine system into high-pressure and low-pressure sections, driving separate propeller-shafts, has already been remarked upon.

(m) THEORY OF TURBINE-ACTION.—We shall now take up the theory of the turbine, following the lines suggested in § 62.

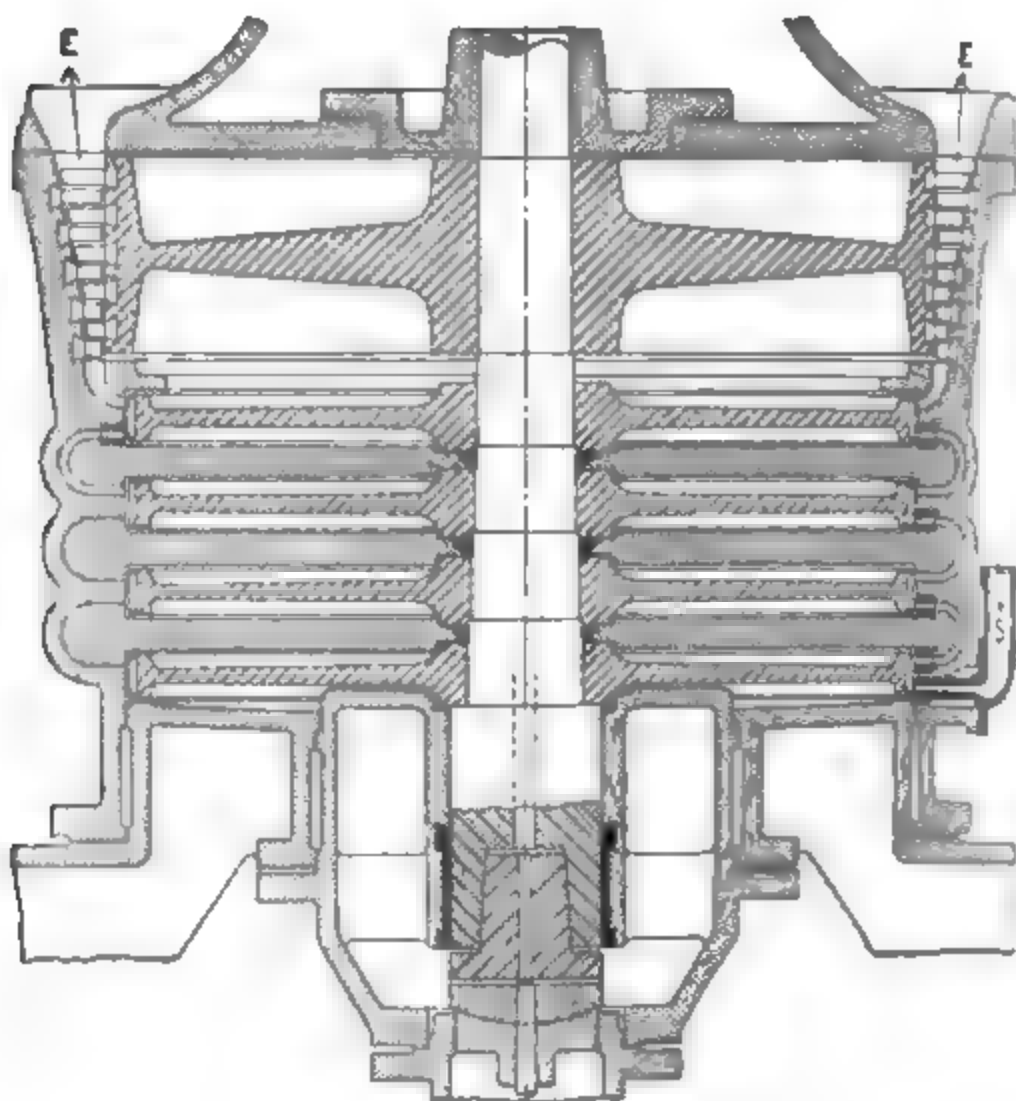


FIG. 631.—The Union Turbine.

the next section the discussion is purely mechanical, and the simplest ideal conditions are imagined to exist. Then in § 73, after the mechanical principles have been established, the thermodynamic questions involved in energy-transformation and in the secondary wasteful actions will be considered. The limits of space available will permit only a very general treatment of this latter field, which is both very extensive and of great practical importance; here



again we shall work for principles and methods, rather than for actual results. For one thing, not enough experiments have been made (and published) for the basis of a close-fitting theory; for another, a full presentation of even the work that has been done along this line would cover a great deal of space.

## § 72. Mechanics of the Ideal Steam-action.

(a) **IMPULSE OF THE JET.**—The first problem is to find the force which a jet of fluid exerts when it impinges upon a restraining surface, or the reaction which it exerts when leaving a confining vessel or channel. This force is called the impulse of the jet, and may be defined as the constant free force which, if it were to act upon the substance of the jet, would continually produce (or destroy) the steam-velocity  $V$ .

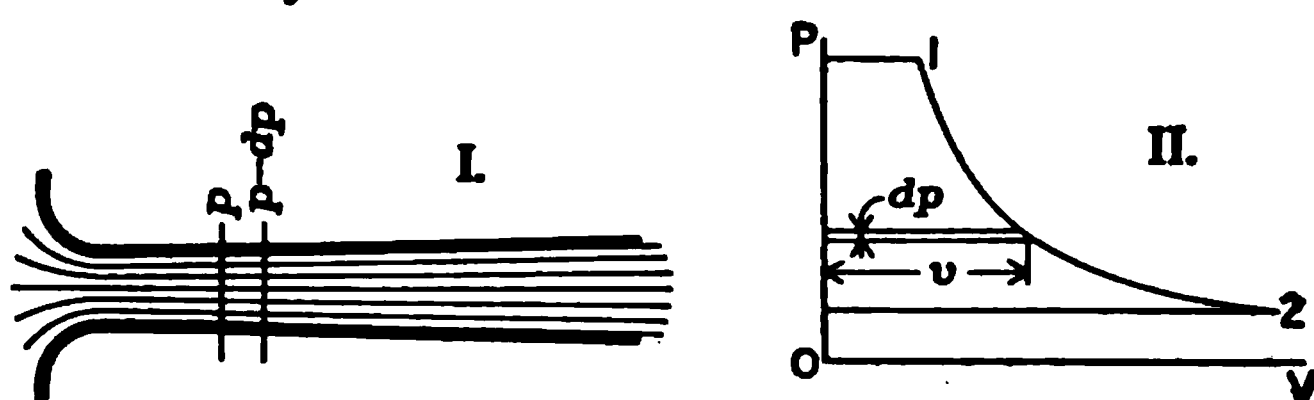


FIG. 632.—Generation of Kinetic Energy.

Really, of course, the steam is accelerated by an unbalanced internal pressure (not of constant intensity), as indicated in Fig. 632 I.; but the conditions of the process are such that quantitative relations must be calculated in terms of energy, rather than directly between force and mass. Fig. 632 II. shows that the element of work is  $v dp$ , and that the total kinetic energy due to pressure-drop from  $p_1$  to  $p_2$  is equivalent to the integral of  $v dp$  between the limits  $p_1$  and  $p_2$ —compare Fig. 45. Primarily, however, this energy can be determined much more easily as a thermal than as a mechanical quantity, and that is the method used in §§ 25 and 26 and in § 73.

Since we are at present concerned with mechanics rather than thermodynamics—with the jet as delivered and applied rather than with the detail of its formation—we may substitute for the

real process of velocity-generation the simpler process of equivalent effect outlined by the definition of impulse given above. The value of the impulse can be most easily derived through the quantity known as momentum. Starting with the fundamental relation,  $\text{Force} = \text{Mass} \times \text{Acceleration}$ , or

$$F = MA, \quad . . . . . (401)$$

and multiplying both sides of the equation by  $t$ , we have

$$Ft = MA t = MV. \quad . . . . . (402)$$

That is, when a free force  $F$  acts upon a mass  $M$  through the time  $t$ , and generates the velocity  $V$  from an initial state of rest, the product of force by time equals the product of mass by velocity, which latter is called momentum. This is very elementary mechanics, but is of such vital importance in the present connection that it is worth repeating with emphasis.

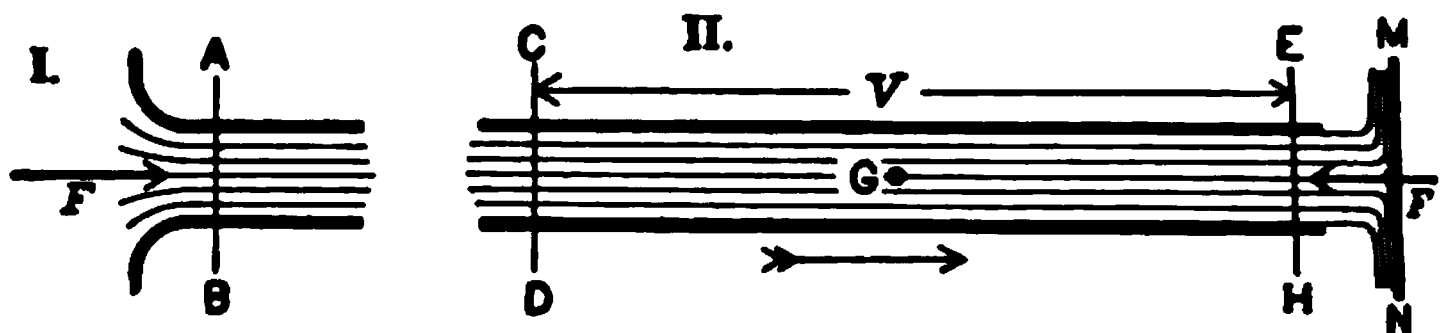


FIG. 633.—Impulse and Reaction of a Jet.

Now consider Fig. 633 I., where the fully formed jet, passing the cross-plane  $AB$ , has the velocity  $V$ , the sectional area  $a$ , and the specific weight  $w$  (pounds per cubic foot). In a time  $t$  the total weight  $Wt = waVt$  will issue, having been raised from zero velocity to  $V$  by the action of the force  $F$  through this same time. Here  $W = waV$  is the weight discharged per second, or the rate of flow. Putting these quantities into (402) we get

$$Ft = \frac{waVt}{g} V = \frac{WtV}{g};$$

or

$$F = \frac{waV^2}{g} = \frac{WV}{g}. \quad . \quad . \quad . \quad . \quad . \quad (403)$$

In words, the force  $F$  is proportional to the density of the fluid and to the square of its velocity; or for a given rate of flow (in weight per unit of time) the impulse is directly as the velocity.

Another way of arriving at the same conclusion is to go back to the elementary principle that when a body, initially at rest, has attained the velocity  $V$  under acceleration by a constant force  $F$ , the distance travelled during the operation is the mean velocity  $1/2 V$  by the time  $t$ . Then the force  $F$ , in acting through the distance  $1/2 Vt$ , does the work that is stored as kinetic energy. Considering what happens to the jet in one second, we have the energy equation

$$F \times \frac{V}{2} = \frac{WV^2}{2g}, \quad . \quad . \quad . \quad . \quad . \quad (404)$$

which reduces at once to the expression in (403).

Something like this latter action is pictured in Fig. 633 II., where the jet impinges upon a flat plate MN, and its energy of forward movement is continually being absorbed or transformed. The mass to be discharged in one second is shown as included between the planes CD and EH, which are separated by the distance  $V$  feet. At the beginning of the particular second under consideration, the plane EH just touches MN; and the center of mass  $G$  being then at the distance  $1/2 V$ , this is the average distance through which the resistance  $F$  will act upon the mass  $W/G$  in bringing it to rest.

At I., in Fig. 633, the force  $F$  is represented as the impulse upon the jet; against the confining vessel is exerted an equal and opposite reaction. Similarly, at II. the force shown is the reaction of the plane MN upon the jet, while the latter at the same time exerts the positive impulse  $F$ , toward the right, upon MN.

(b) DEFLECTION OF A JET.—In Fig. 634 a flowing jet or stream is depicted as entering at A and leaving at B a frictionless channel of uniform curvature and cross-section. The velocity remains

constant in intensity, and the action of deflection is that of simple centripetal motion. The reaction of the stream reacts the centrifugal force of the stream.

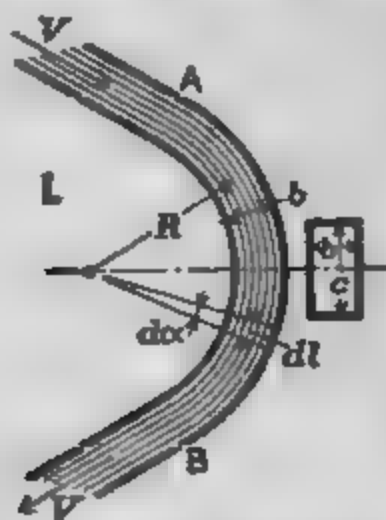


FIG. 634.—Deflection of a curved stream.

The essential dimensions and symbols are:

$a$  = area of cross-section of channel, having the width  $b$  and the depth  $c$ ;

$R$  = radius of center-line of stream;

$\alpha$  = angle between any pair of radii;

$l$  = length measured along curved center-line;

$v$  = specific volume of fluid in jet;

$w$  = weight per cubic unit =  $1/v$ ;

$V$  = velocity of flow;

$A$  = acceleration;

$W$  = weight of fluid discharged per second;

$F$  = impulse of jet.

Consider an element of the stream between two radial planes at the angular distance  $d\alpha$ .

$$\text{Length} = R d\alpha; \quad \text{volume} = a R d\alpha; \quad \text{weight} = w a R d\alpha$$

The centrifugal force of this element is  $m l A$ .

$$f_c = \frac{w a R d\alpha}{g} \frac{V^2}{R} = \frac{w a V^2}{g} d\alpha$$

This equation sets forth the very important result that the total force required to deflect the jet through an angle  $\alpha$  is equal to the impulse of the jet multiplied by the angle of deflection, the latter being expressed in absolute angular measure, or with the radian as unit. This force, got by integrating Eq. (405) from zero to  $\alpha$ , as indicated at II. in Fig. 634, is uniformly distributed along the curved path from A to B.

Another fact of interest is, that for a given set of jet-conditions (including the area  $a$ ) the total deflecting force required, or the reaction of the jet upon the curved surface, is independent of the radius of curvature. Referring to the first expression for  $f_c$  in Eq. (405), we see that to make  $R$  longer increases the mass involved, by making the channel longer for a given angle, but at the same time decreases the centripetal acceleration  $V^2/R$ , so as to keep the product the same. The centrifugal pressure per unit of area changes, however, because the restraining area varies with the mass involved. Thus in Fig. 634 I. the outer surface of the element can be taken as  $cRd\alpha$ —disregarding the fact that the outer radius is really  $(R + \frac{1}{2}b)$ : then dividing this into  $f_c$  from (405), we get

$$p_c = \frac{wbcRd\alpha}{gcRd\alpha} \frac{V^2}{R} = \frac{wb}{g} \frac{V^2}{R} \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad \cdot \quad (406)$$

as the extra pressure due to centrifugal force.

If, however, the width  $b$  also varies with  $R$  in constant ratio, as when two channels of different absolute size are geometrically similar in shape, then  $p$  will remain constant, but the total centrifugal force  $F_c = \Sigma f_c$  will increase with  $R$ .

**Example 1.**—Let a steam-jet at 75 lbs. absolute pressure, with the velocity 1663 ft. per sec. and the specific volume 5.50 cu. ft. (see Table 24 A, 150 lbs. to 75 lbs.), flow in a channel  $1/2''$  wide by  $1''$  deep, with the mean radius  $2''$ . What is the impulse of the jet and what the centrifugal pressure on the guiding surface, disregarding effects of friction?

Here  $a = 0.5$  sq. in.,  $= .00347$  sq. ft.

The rate of flow is  $W = \frac{aV}{v} = 1.050$  lbs. per sec.

The impulse is  $F = \frac{1.05 \times 1663}{32.16} = 54.3$  lb

In applying Eq. (406) to find centrifugal force, not to get feet and inches mixed. With  $b$ , centrifugal force, as given in (405) becomes

$$f_c = \frac{w}{g} \frac{bc}{144} \frac{R d\alpha}{12} \frac{12V^2}{R} = \frac{wb}{144g} \frac{V^2}{R}$$

while the area involved, when we want to  $cRd\alpha$  directly: then (406) takes the form

$$P_c = \frac{wb}{144g} \frac{V^2}{R}$$

For this problem, noting that  $w = 1/5.50$ ,  $V = 1663$  ft./sec.

$$P_c = \frac{0.5 \times 1663 \times 1663}{5.50 \times 144 \times 32.16 \times 2} = 27.15$$

Using the larger actual outer surface of 2 1/4" instead of 2", this pressure changes

$$27.15 \times \frac{2}{2.25} = 24.13 \text{ lbs./sq. in.}$$

An obvious conclusion from this example is considerable crowding of the stream again, corresponding variation in the pressure with

(c) DRIVING FORCE ON THE VANE.—force exerted by the jet upon a curved vane is to find the resultant, in a certain direction distributed along the vane. The problem where the tangential force  $F_T$  is the resultant direction of motion of the vane or bucket, elementary force  $f_c$ , we get the driving component the axial component  $f_A = f_c \sin \alpha$ , the latter axis in the usual type of axial-flow turbine two integrations, one on each side of the

405) we have

$$f_T = \frac{u \dot{V}}{g} \cos \alpha d\alpha; \quad . . . . . (407)$$

then, for the resultant force,

$$\begin{aligned} F_T &= F \left( \int_0^{\alpha_1} \cos \alpha d\alpha + \int_0^{\alpha_2} \cos \alpha d\alpha \right) \\ &= F (\sin \alpha_1 + \sin \alpha_2), \quad . . . . . (408) \end{aligned}$$

$\dot{V}$  being the impulse of the jet, as appears from (403).

For the resultant axial force  $F_A$ , a similar deduction gives

$$F_A = F (\cos \alpha_1 - \cos \alpha_2); \quad . . . . . (409)$$

obviously, it is highly desirable that this force be made zero, otherwise there would be a dynamic end-thrust on the rotor.

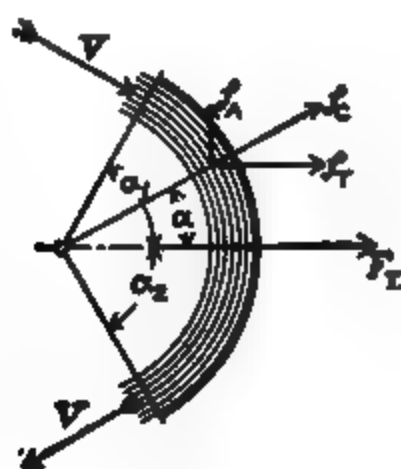


FIG. 635.—Resolution of Centrifugal Force.

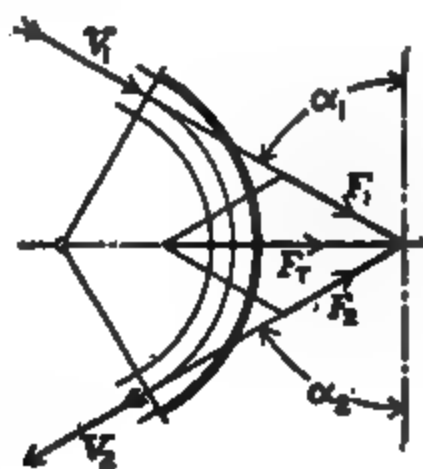


FIG. 636.—Combining the Impulses.

The form of Eq. (408) suggests at once the simple method set forth in Fig. 636. Considering the jet as exerting a positive impulse  $F_1$  at entrance and a negative impulse or reaction  $F_2$  at exit, we have only to combine these forces, or their rectangular components, to get  $F_T$  and  $F_A$ . Since impulse is proportional to velocity, this gives a very easy and convenient method for the solution of problems in force-action on the vanes of a turbine.

The velocity-diagrams given in Fig. 636 are worked out by the method already used in § 70. If the relative velocity of entrance  $V_1$ , we neglect the absolute jet-velocity  $V$  the reversed turbine velocity of the nozzle with reference to the absolute discharge-velocity  $V_0$ . The direct must be the exit-velocity  $V_2$ .

(d) TYPES OF VANE-ACTION.—The diagrams at I. in Figs. 637 and 638 are presented to develop a clear idea of the meaning of the reaction action as used technically in the present case the vanes are so formed that the reaction is perpendicular to the line of motion at entrance is effective to produce driving action. This abstraction is very imperfect, as appears from the velocity  $V_0$ . Another way of expressing it is to say that the resultant  $F$  and the work done are relatively small because  $F_1$  and  $F_2$  have large components in the direction of motion. Very evidently this type of action is obtained by making the vanes symmetrical with respect to  $AB$ , thereby getting the well-known form of the reaction turbine etc. In other words, what is called a reaction turbine is driven equally by impulse and by reaction.

But while the scheme outlined in Fig. 637 is of Fig. 638 is entirely so. Receiving the steam (the line of motion) with the velocity  $V_1$ , the turbine converts it to  $V_2$  and discharges it at a wide angle. This leads to the force-diagram III., where the resultant  $F$  or  $F_T$  is large and is right along the line of motion. It appears then that while the "impulse" turbine the "reaction" turbine can get along with the same distinction lies in the fact, stated in § 71 (the reaction is generated wholly in the nozzles, in the fixed and moving vanes. The characteristic velocity diagrams 639 and 641 result from this underlying understanding of what lies back of the technical action to the ordinary nomenclature, and



be drawn in that the effective exit-reaction in an impulse turbine is due wholly to deflection of the jet, while that in the reaction

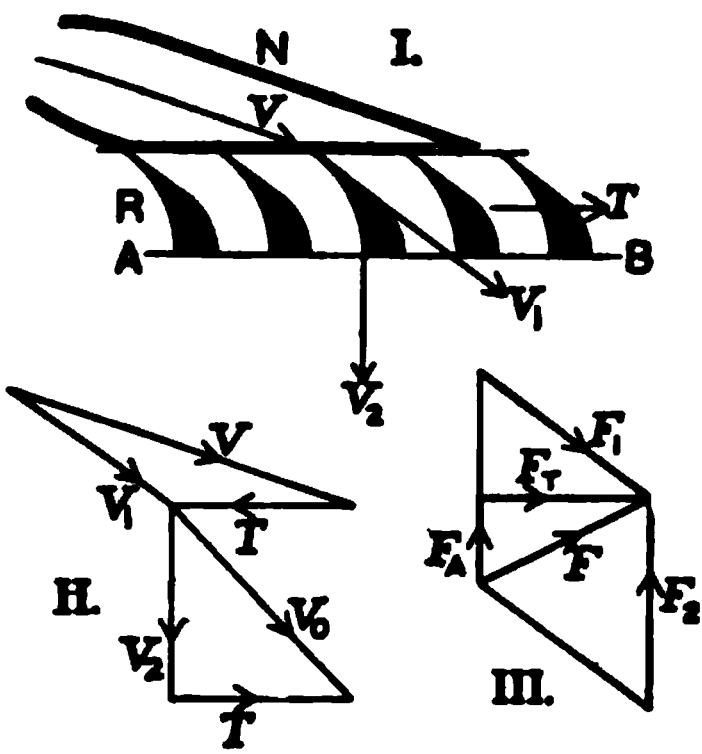


FIG. 637.—Driving by Impulse Only.

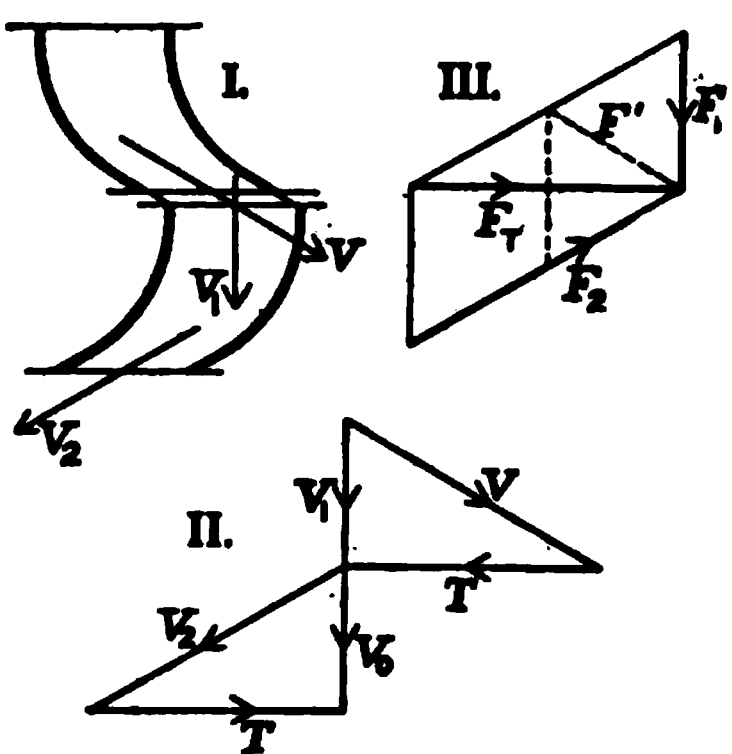


FIG. 638.—Driving by Reaction Only.

turbine is due to both deflection and linear acceleration. Comparing  $F_T$  with  $F'$  on Fig. 638 I., we see the gain due to acceleration within the wheel, over an impulse-vane with the same entrance-velocity.

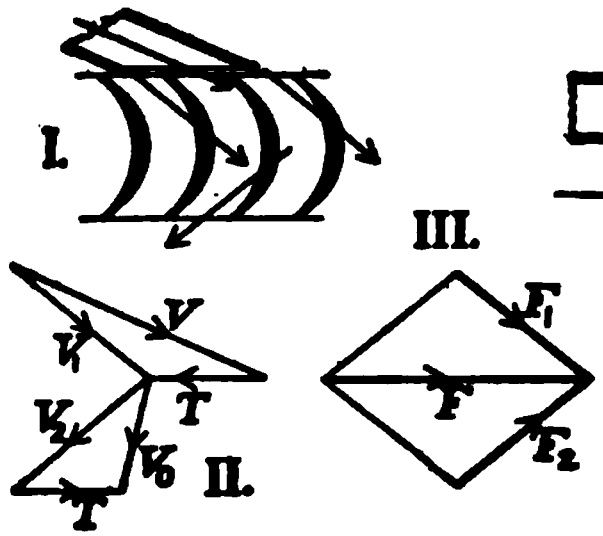


FIG. 639.—The Vane, with Side Admission.

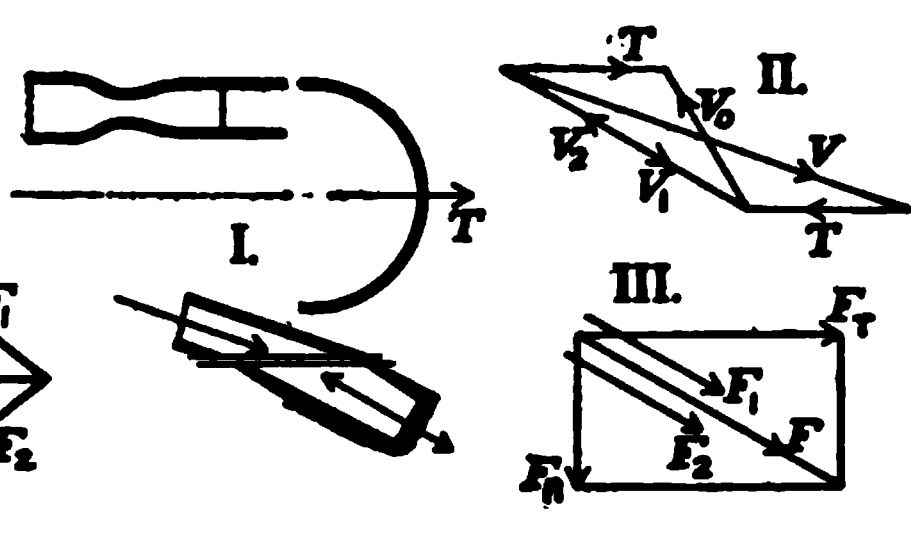


FIG. 640.—The Bucket, with Tangential Admission.

After the discussion just given, the actual profiles and their diagrams of velocity and force, as set forth in Figs. 639 and 640, ought to be self-explanatory. In Fig. 640, for the wheel with tangential admission as in Fig. 624, the plane of the diagrams at II.

and III. is that of the lower view at I, perpendicular to the axis, in which the wheel diagram doubles back on itself, because of the velocity  $V_1$  reversed. It is necessary, of course, to show the steam in the radial direction, since the steam must flow in that direction. The forces  $F_1$ ,  $F_2$ , and  $F_3$  are shown in the same line in their diagram, but are here shown in a different representation. Whereas with side view the driving-force, the same thing is shown in the mission there is a non-effective component pressure in the bearings, but which can be eliminated by admitting steam to diametrically opposite



FIG. 641.—Diagrams for the Reaction Turbine.

The reaction vanes in Fig. 641 differ from those in Fig. 638 in that they provide for a smaller clearance. Since it is always desirable that the clearance be no greater than is absolutely necessary, the clearance is made a little less than the projection of  $V$ ,  $g$ . Then the vane-profiles are made to fit the clearance in any case.

(c) WORK ON THE VANES.—The effect of the reaction vanes is to produce a force  $F$  (called  $F_T$  in the preceding discussion) which has the velocity  $T$ ; then the power, or work, is

$$P = FT \text{ ft. lbs. per sec.}$$

We shall now apply this equation to several typical cases. We shall still adhere to the assumption that there are no losses by friction or by other

the figures immediately following, the velocity diagrams (from which the impulsive forces may be directly determined) are changed to a rather more compact form than that used heretofore.

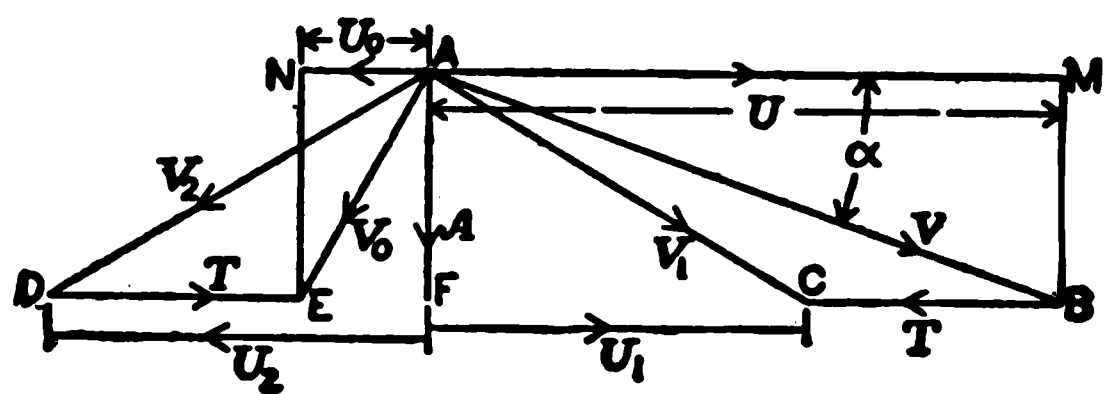


FIG. 642.—Velocity Diagram for the Impulse Element.

In Fig. 642, for instance, all the steam-velocities are laid out from the point A as an origin: AB is the initial absolute velocity  $V$ , AC the relative entrance-velocity  $V_1$ ; the exit-velocity AD or  $V_2$  is  $V_1$  reversed symmetrically; and AE is the final absolute velocity, or  $V_0$ . For a discussion of work-performance we are concerned, however, not with any total velocity  $V$  so much as with its component  $U$  in the direction of motion, or along  $T$ ; the effective impulses at entrance and exit being proportional to the velocities  $U_1$  and  $U_2$ , according to Eq. (403). With the relation

$$V^2 = U^2 + A^2, \quad . . . . . (411)$$

and with the axial component  $A$  remaining constant throughout the successive transformations which take place, we see that changes in the kinetic energy of the steam-current are represented and measured by changes in the value of  $U^2$ .

Now for the impulse turbine, as represented by Fig. 642, and with the several velocities as there designated, the fundamental expressions are

$$U_1 = U - T \quad \text{and} \quad U_2 = U_1, \quad . . . . . (412)$$

the latter equation embodying the condition of symmetrical reversal, and being subject to modification in the actual case. The effective impulses are now

$$F_1 = \frac{W}{g} U_1 \quad \text{and} \quad F_2 = \frac{W}{g} U_2,$$

and the work-rate is

$$P = \frac{W}{g} T (U_1 + U_2) = 2 \frac{W}{g} (UT - T^2). \quad (413)$$

Examining this for the maximum value of  $P$ , in the usual manner, we get

$$\frac{dP}{dT} = 2 \frac{W}{g} (U - 2T), \quad (414)$$

which becomes zero when  $T = 1/2U$ , thus proving the oft-stated principle that the velocity of the vanes should be one-half the effective velocity of the steam-jet for maximum efficiency. The greatest work-rate is now

$$P_1 = 2 \frac{W}{g} \times \frac{1}{4} U^2 = \frac{1}{2} \frac{W}{g} U^2; \quad (415)$$

or, as it should be, the full kinetic energy available in the weight  $W$  of steam that passes in one second.

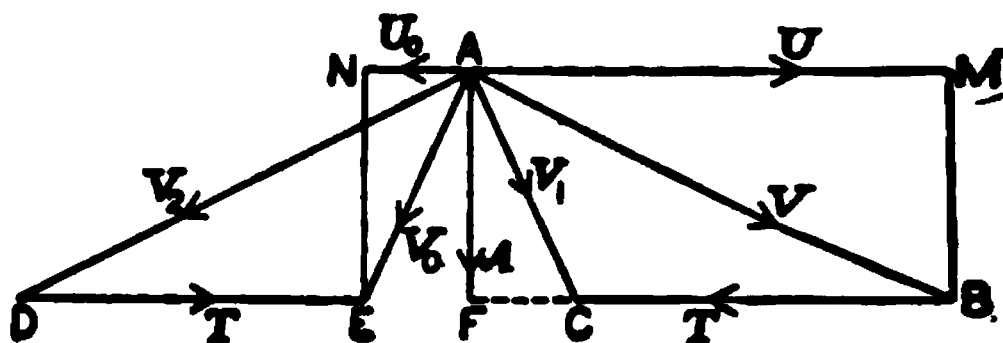


FIG. 643.—Velocity Diagram for the Reaction Element.

From the similar representation of the velocities in a reaction-turbine element, Fig. 643, we get the relations

$$U_1 = U - T, \quad U_2 = U, \quad (416)$$

the second implying similarity between fixed and moving vanes. Then

$$F_1 + F_2 = \frac{W}{g} (2U - T),$$

and

$$P = \frac{W}{g} (2UT - T^2). \quad (417)$$

This is greatest when  $U = T$ , in which case the effective driving force is wholly due to reaction, since  $U_1$  will be zero or the steam will enter the vanes at right angles to  $T$ . The maximum value

of  $P$  is now

$$P_1 = \frac{W}{g} U^2; \quad . . . . . (418)$$

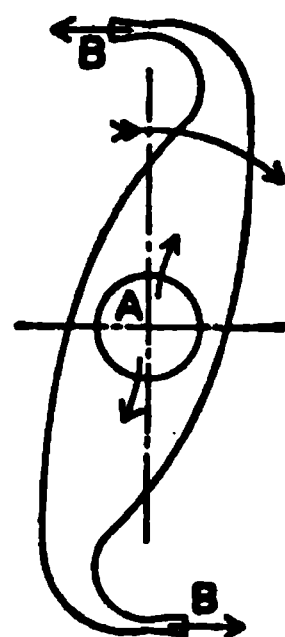
but this  $U^2$  represents only half of the energy of one complete stage, since  $U$  is generated twice, first in the fixed vanes, again in the moving vanes. The total kinetic energy generated and absorbed is properly expressed by

$$P_1 = \frac{1}{2} \frac{W}{g} (2U^2), \quad . . . . . (419)$$

this  $(2U^2)$  being equivalent to the  $U^2$  in Eq. (415).

(f) THE REACTION PRINCIPLE.—The fact has just been brought out that a reaction turbine transforms a certain amount of pressure-volume energy (one stage) in two parts, twice developing a steam-velocity 0.707 times as great as an impulse turbine would get in a single operation: and a good way to look at the performance of the reaction turbine is to consider that the function of the fixed element is simply to deliver steam to the vanes at their own velocity; whereupon the acceleration of this steam backward, by and within the moving vanes, produces the effective driving force.

This idea is best illustrated by the reaction wheel, which it is appropriate to consider briefly at this point, and of which the essential form is shown in Fig. 644. Steam is admitted to the interior of the rotor at the axis, flows out along the hollow arms, and is discharged through suitable nozzles at B, B. The reaction of the jet is the driving force, and maximum efficiency is secured when the velocity of the nozzles just equals that of the jet, so that the absolute velocity of discharge is zero. Here again, as in the turbine, half of the energy due to the pressure-drop is used in getting the steam-mass up to the velocity which it has at the end of the hollow arm, just within the nozzle (on account of the rotation of the wheel); and the other half accelerates the steam backward.



This device has been applied by a number of inventors, but two

reasons lie against its effective use: the chief is the high velocity due to single-stage expansion; the other, the practical difficulty of avoiding leakage (or else excessive friction in the joint) where the steam enters the rotor—the stuffing-box problem being greatly aggravated by the fact that the steam has full pressure at the point.

It is of interest to note that for a stage of the same energy-value the reaction turbine must move faster than the impulse type. If the energy available will generate the effective velocity  $U_1$ , this will be the velocity realized in the impulse nozzles, and the best vane-speed will be  $T_1 = 1/2 U_1$ . But with the reaction arrangement, the steam-velocity will be  $U_R = 0.707 U_1$ , and  $T_R$  must then have this same value, or must be 1.414 times as great as  $T_1$ . Actually, of course, the reaction turbine is usually made with the large number of smaller stages.

(g) VARIATION IN THE RUNNING-SPEED.—It will be noted that neither Fig. 642 nor 643 is drawn for the case of ideal maximum efficiency, but rather for the conditions likely to be found in practice, where the vane-speed  $T$  is made as low as is consistent with reasonably good working. To see the effect of thus lowering  $T$  from the ideal value, we discuss Eqs. (413) and (417) as follows:

For the impulse turbine      For the reaction turbine

$$P = 2 \frac{W}{g} (U - T) T; \qquad P = \frac{W}{g} (2U - T) T.$$

In both cases, let  $T = nU$ , and put the expression into the form kinetic energy  $\times$  a function of  $n$ ; this gives

$$P = \left( \frac{1}{2} \frac{W}{g} U^2 \right) (1 - n) 4n; \quad P = \left( \frac{W}{g} U^2 \right) (2 - n) n. \quad (41)$$

For the first case, let  $m_1 = 4n_1(1 - n_1)$ , and evaluate for  $T$  varying in twentieths from zero to  $1/2 U$ ; for the second, let  $m_2 = n_2(2 - n_2)$ , and go by tenths from zero to  $U$ . The common results are

$n_1 = 0$	0.05	0.1	0.15	0.2	0.25	0.3	0.35	0.4	0.45	0.5
$n_2 = 0$	0.1	0.2	0.3	0.4	0.5	0.6	0.7	0.8	0.9	1.0
$m_1 = m_2 = 0$	0.19	0.36	0.51	0.64	0.75	0.84	0.91	0.96	0.99	1.0

It is evident that the velocity  $T$  can be lowered to 70 or 75 per cent. of the ideal value without serious loss of effect. Further, we can see that these values of  $m$  might equally well have been gotten from Fig. 642 or 643 directly, by noting how the effective component  $U_0$  of the residual velocity  $V_0$  will vary with  $n$  or  $T$ .

An important assumption underlying the above table is that the vanes be changed in form, with the running-speed, so as to get full effect from the conditions existing in any case: and this leads us to the next matter to be taken up.

(h) VANE FORM AND THE EFFECT OF SPEED-CHANGE.—The proper function of a set of curved vanes or guides in a turbine is, to receive a current of steam without shock, to change its direction without the formation of eddies, and to discharge it in a desired direction. We will now consider some of the simpler questions involved in the performance of this function.

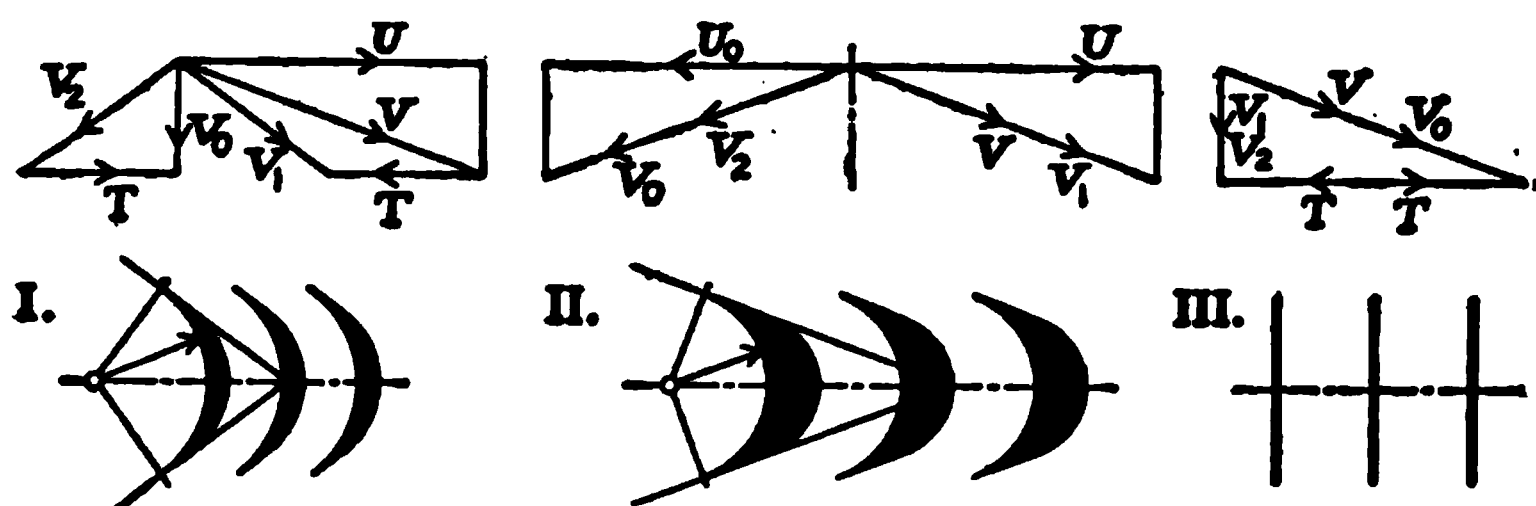


FIG. 645.—Vanes to Fit Various Speeds.

The first and most elementary problem, that of accommodating the shape of the vane to a proposed velocity diagram, is illustrated by Fig. 645. In Case I. the conditions are those for maximum efficiency in an impulse turbine, the vane-velocity  $T$  being half of the effective entrance-velocity  $U$ : the concave profiles are arcs of circles, made tangent to the lines of direction of the relative velocities  $V_1$  and  $V_2$ ; the convex profile is made up of a smaller arc and two tangents. If the channel between two vanes is to have an approximately constant width, the vane must, of course, be thickened toward the middle. This effect is much exaggerated at II., where the vanes are proportioned so as to receive and symmetrically reverse the full entrance-velocity, when  $T=0$  or the vanes are





happen if the turbine were "running away." It is seen that at entrance the current will exert a retarding impulse upon the back of the vane. Although this condition is highly favorable to eddy-losses, we for the present show  $V_2$  as equal to  $V_1$ , and tangential to the vane at exit: then the final velocity  $V_0$  has a direction approaching that of  $V$ , instead of pointing backward.

Cases II. and III. in Fig. 646 emphasize the requirement stated in (e), under Eq. (411), for the relations there developed; namely, that the progressive or axial component of the several entrance- and exit-velocities must be the same if change in kinetic energy is to be represented by change in the tangential velocity  $U$  (or in  $U^2$ ). Here we see that as soon as the condition of symmetrical reversal from  $V_1$  to  $V_2$  fails to be realized,  $U_0$  ceases to be a true criterion of the residual energy, and we must go back to the comparison of  $V_0^2$  with  $V^2$ .

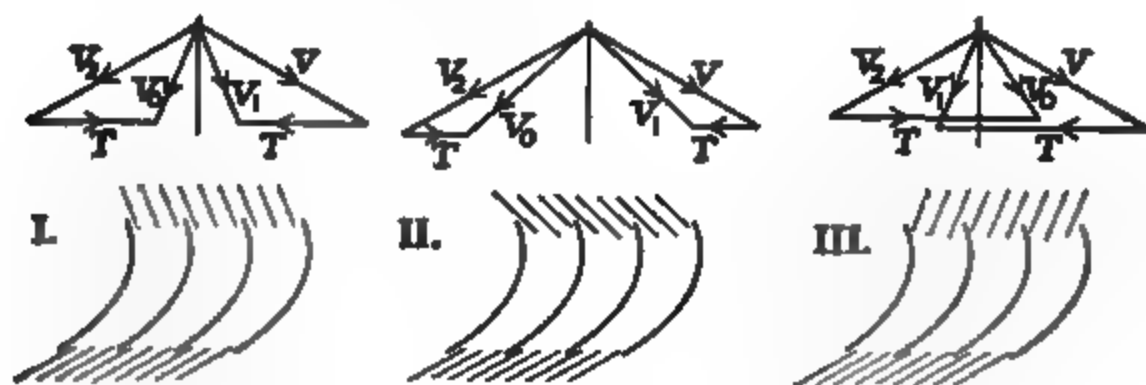


FIG. 647.—Speed-change with the Reaction Element.

A similar discussion of the reaction turbine is represented in Fig. 647: I. shows vanes for the normal speed  $T=0.75 U$ : II., for lower speed, shows that the impulse at entrance becomes of relatively greater importance as the speed drops, even if we assume that none of the kinetic energy at entrance is lost within the vane, but that all of this plus the effect of the pressure-drop is present at exit. The conditions at excessive speed are closely analogous to those in Fig. 646 III., but are here even less favorable at entrance: and in Fig. 647 III. account is taken of the shock and eddy losses, in that  $V_2^2$  is less than  $V_1^2$  plus the same pressure-drop effect that is shown at I.

The question of the effect of speed-change in a many-stage

turbine is very complex: but we can see that if the cross-section of the total steam passage for a certain running-speed with a certain pressure-drop, slower running will cause a loss of energy rapidly, while over-speeding will probably cause a gain. The argument is that with under-speeding the steam has increased velocity in and out of the way more rapidly; but if the speed is too high—the latter being less than the speed for maximum energy—more energy will be taken out of the steam by the velocity to carry it along. This loss of energy is due to eddy effects, especially because the steam does not flow smoothly.

A brief discussion of the secondary action will be found in § 73 (k).

(i) CHANNEL FORM AND AREA OF CHANNELS.—In the usual type of turbine, that with axial flow and side admission, the channels for the passage of steam are an annular space, which changes in diameter according to the requirement for effective flow. It has, at any critical point, an actual velocity  $V$  and a component-velocity  $A$  (see Figs. 642, 643, 644), to the axis or normal to the line of vane velocity, as also the direction of the channel walls and the rectangular reference-lines.

To understand the effect of this obliquity of the walls where the vanes in  $R$  are made straight, compare the actual vanes would have at entrance—being curved—with the velocity-diagram II. Letting  $b_0$  represent the width of the channel in the circumference, and  $b_1$  that the effective width is much less, having

$$b = b_0 \sin \alpha \quad \text{or} \quad b_1 = b_0 \sin \alpha$$

in other words, making the walls helical and the channel by an amount which increases with the angle to the axial direction.

In the turbine, the relation between



shown at II. in Fig. 648, is the same as that between the channel-widths (in inverse order) for

$$V = \frac{A}{\sin \alpha} \quad \text{and} \quad V_1 = \frac{A}{\sin \alpha_1} \quad . \quad . \quad . \quad (422)$$

Consequently the flow-capacities are the same, since

$$Vb = V_1 b_1 = Ab_0 \quad . \quad . \quad . \quad . \quad (423)$$

The last expression,  $Ab_0$ , is then the criterion by which to measure the effective area for the passage of steam.

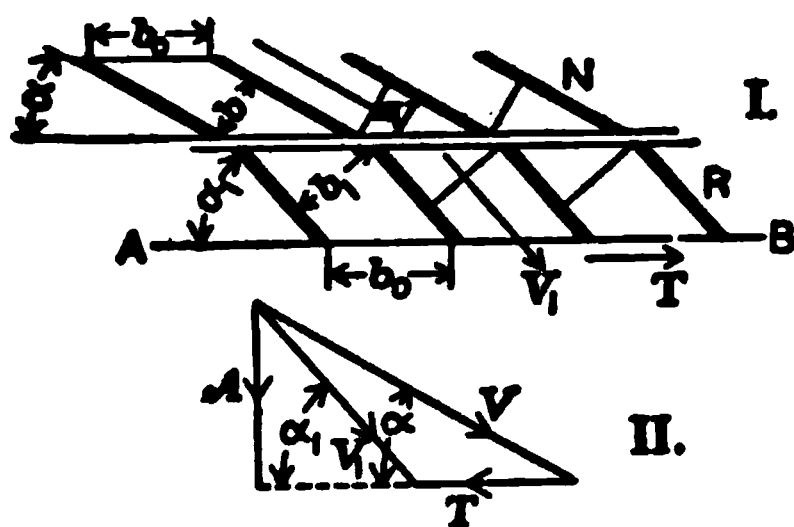


FIG. 648.—The Oblique Channel.

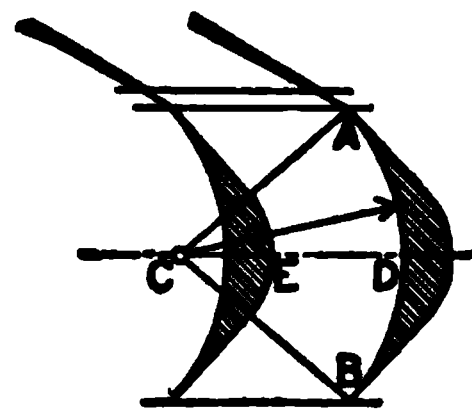


FIG. 649.—Channel of Constant Width.

The preceding discussion determines relations which in the actual case exist at entrance to and exit from the channels between curved vanes. For the body of such a channel the conditions which give constant width are set forth in Fig. 649, where the curved portion of the passage is included between the radial lines CA and CB, the same center C being used for the arcs through D and E. The straight lines (tangents) which form a large part of the outer profile of the vane-section have, of course, the inclination of the velocity  $V_1$  as in Fig. 648, making the angle  $\alpha_1$  with the line CD.

An interesting conclusion from Eq. (423) is that if in an arrangement like Fig. 623 the vanes are brought to a sharp edge and formed as in Fig. 649, the diminution of obliquity will compensate for the diminution of velocity due to abstraction of energy—this on the assumption that the velocity changes are according to a diagram like Fig. 650, where  $A$  remains constant. Only for the purpose of accommodating extra losses of progressive velocity, as by friction

and eddies, need the channels be increased in radial depth with the pressure-stage.

(j) **WORK-RELATIONS IN THE COMPOUND TURBINE.**—The amount of pressure-drop from stage to stage in any compound turbine depends primarily upon the manner of variation of the total cross-section for the passage of steam—a question which will be taken up presently. A point to be brought out here, based on whirling mechanical considerations, is that the amount of kinetic energy properly to be generated and absorbed in each stage depends upon the velocity of the vanes. With a rotor having several diameters, as in Figs. 620 and 629,  $T$  is much larger in the lower than in the higher stages: but if the efficiency in energy-absorption is to be the same throughout, there must be nearly a constant ratio between  $V$  and  $T$ . We see then that the lower stages must be larger, in energy-value, than the upper stages; and, as a simple relation, this value of the stage must vary directly as the square of the varying diameter, or as the square of the corresponding velocities involved.

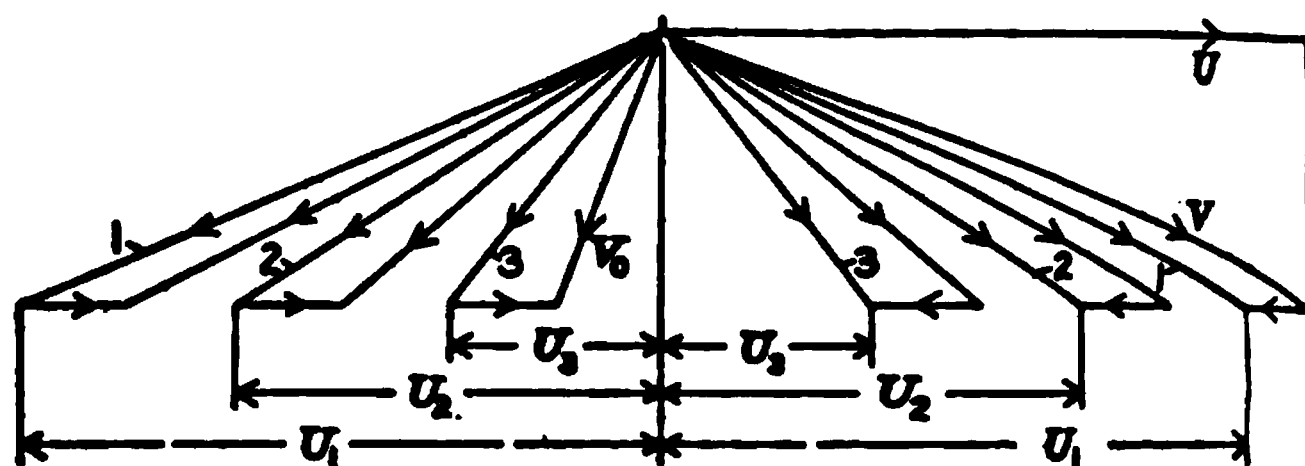


FIG. 650.—Velocity-diagram for a Three-impulse Stage.

Fig. 650 illustrates the relation between the quantities of work in the several velocity-stages of a multiple-impulse turbine, this velocity-diagram being constructed in the usual manner, for a three-impulse element like Fig. 623. In the figure the relative velocities of entrance and of exit are marked by the numbers 1, 2, and 3 for each set of vanes—compare Fig. 74. Following the method of Art. (e), and dropping for the time the factor  $W/g$  in Eq. (403), we have the following expressions for the driving-force in the respective stages:

$$F_1 = 2(U - T); \quad F_2 = 2(U - 3T); \quad F_3 = 2(U - 5T). \quad (424)$$

Of course, the velocity-factor  $T$  in  $P_u = F_u T$  is the same all through, so that the work-rates are proportional to the impulsive forces. If, for instance,  $T$  is one-seventh of  $U$ , as in the figure, the quantities of work done by the steam upon the vanes are as 6, 4, and 2 in the three velocity-stages.

To show the identity, under the conditions of simple theory, of the above example with the equivalent single-impulse element, we first add the three driving-forces in (424), then multiply by  $T$  to get the power developed, the results being

$$F = 6(U - 3T); \quad P = 2(U - 3T) \times 3T.$$

Since a single-impulse stage with the same limits  $V$  and  $V_0$  as in Fig. 645 would have the vane-speed  $T' = 3T$ , the desired equivalence is self-evident.

(k) **PATH OF THE JET.**—The absolute path of an element of the steam-current as it passes along the moving vane is a matter of interest, and, in cases like Figs. 619 and 624 III. and IV., of considerable practical importance. Plotting this path is a simple geometrical process, which is shown in Fig. 651 for the side-admission **VANES.**

First of all, the vane-profile  $ABC$  is divided into, say, eight equal parts; then, from either velocity-triangle, the distance that the vane will move on account of  $T$  while the steam travels over one interval with the velocity  $V_1$  or  $V_2$  is found. This is laid off parallel to  $BD$ , the proper number of times from each numbered point on  $ABC$ , and the result is the path  $ADE$ , which is tangent to  $V$  and  $V_0$  at  $A$  and  $E$  respectively. For a tangential bucket, as in Fig. 624, the method would be essentially the same: in the reaction turbine it would be necessary to know how the steam is accelerated within the bucket; but in peripheral admission there is no need of this determination.

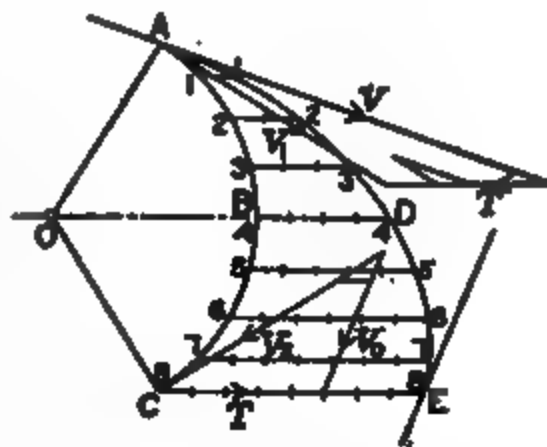


FIG. 651.—Plotting the Path of the Steam-current.

(1) THE LIMITS OF SIMPLE MECHANICAL action have now been reached. We may find the actual performance, using the mechanical and the thermodynamic methods and § 26, and following the lines laid out in §

### § 73. Actual Performance of the Turbine

(a) THE LIMIT OF THE THERMODYNAMIC cycle, or the operation of maximum attainable cycle, represented by ABCE in Fig. 61. For turbine-performance, the entropy-temperature diagram is more serviceable than the pressure-volume diagram of the former, derived from Chapter VI. or as a necessary preparation for what follows, see also on a review of Figs. 79 and 80. Familiarity with the theory in Chapters III. and V. is also assumed.

In § 24, Eq. (116), was derived an expression for the energy represented by the effective area

$$E = q_1 + x_1 r_1 - q_2 - x_2 r_2 + A(l_1 - l_2)$$

This was got by the pressure-volume method only in finding  $x_2$  from  $x_1$  for the turbine cycle. To get the same formula from Fig. 652, we may use the water-heating curve PA, down to a point under PE we have  $q_2$  and under AB the heat rejected since  $x_1$  is here unity. The heat received from feed-water at  $T_2$ , is, then, as in § 9 (d) or

$$Q_1 = q_1 - q_2 + r_1 \quad \text{or} \quad Q_1 =$$

and the heat rejected, under CE, is very nearly

$$Q_2 = x_2 r_2,$$

although it can be, perhaps, more easily obtained from the following formula

$$Q_2 = (a_1 + b_1 - a_2)T$$

The meaning of  $a$  and  $b$ , which are the entropies of heating and of evaporation, taken from the Steam-table, is fully explained in § 13 (c).

Subtracting  $Q_2$  in (427) from  $Q_1$  in (426), we get (425) for  $x_1 = 1.00$ , except that the last term, the water-volume effect, is lacking. This must be taken into account, because the work done by the

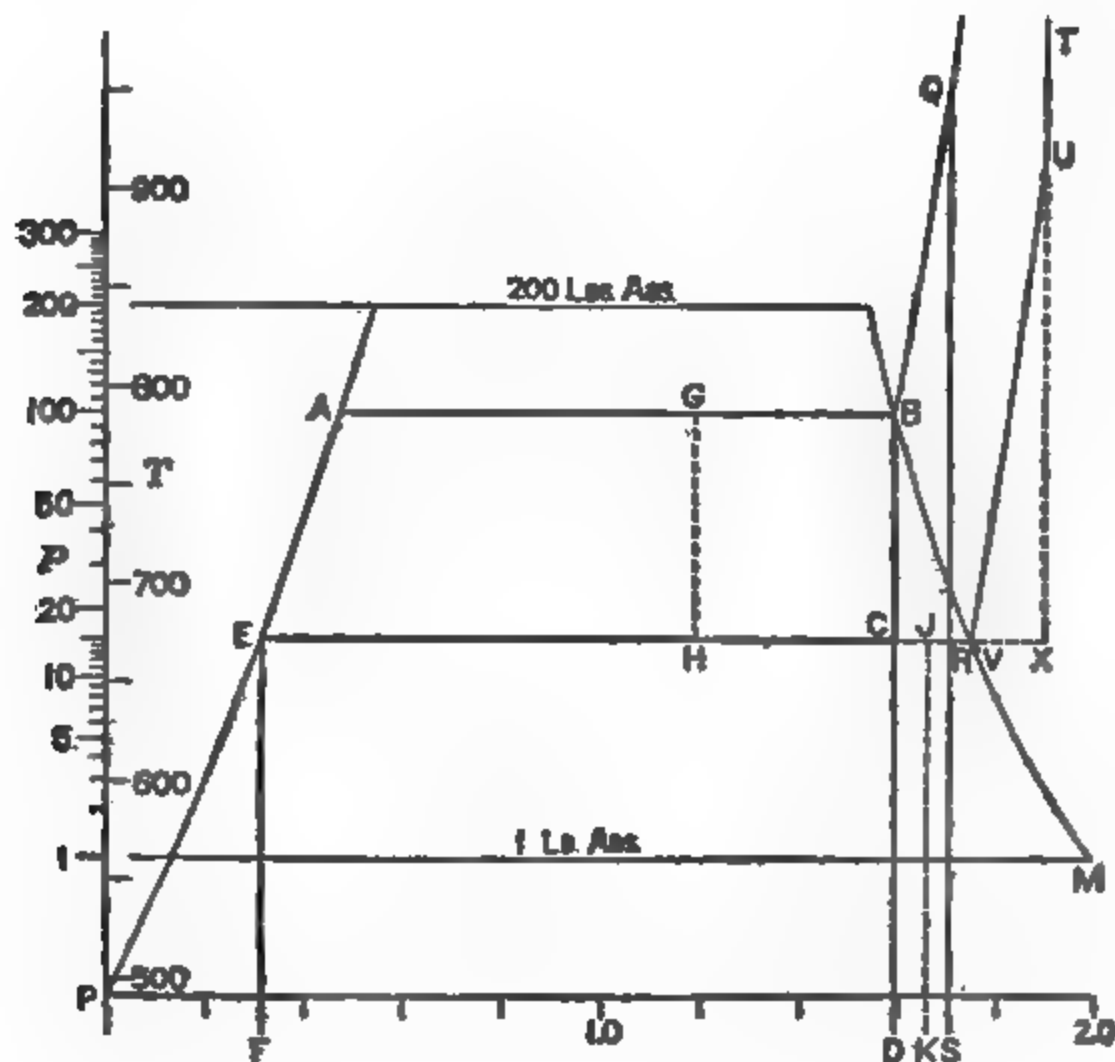


FIG. 652.—Entropy-temperature Diagram of the Rankine Cycle.

feed-pump in forcing water into the boiler reappears at the engine-piston or in the steam-jet; but it must be added to the cycle as something extraneous rather than a part of the main operation itself. It is more strictly correct to change this term to  $A(P_1 - P_2)w_p$ , because any external work of expansion as the water is heated from  $T_2$  to  $T_1$  in the boiler is included in the water-heat ( $q_1 - q_2$ ).

For the area of ABCE, Fig. 652, or for the disposable work ex-

pressed in heat units when the steam  
have

$$\begin{aligned} E &= q_1 + r_1 - q_2 - x_2 r_2 + A(P_1 - P_2) \\ &= H_1 - q_2 - (a_1 + b_1 - a_2)T_2 + (P_1 - P_2) \end{aligned}$$

The latter formula was used in the  
where the values of  $E$  are so closely  
linear interpolation will give sufficiently  
intermediate conditions.

Having this table, it is easiest to find  
the steam is wet or superheated by calcu-  
lating the quantity to be subtracted from or added to  
when the steam contains the fraction  $x_2$ .  
The area ABCE will be diminished by a rectangle  
its height  $T_1 - T_2$  or  $t_1 - t_2$ , so that

$$\Delta E = -m_1 b_1 (t_1 - t_2)$$

If the steam is superheated  $t_s$  degrees  
heat  $c$  (under constant pressure), the total  
of the area DBQS, is

$$h = c_1 t_s.$$

The portion of this heat rejected, under  
entropy CR and the lower temperature  $T$   
Eq. (175)

$$N_s = 2.3026 c_1 \log \frac{T_1 + t_s}{T}$$

Then the area CBQR is

$$\Delta E = + (c_1 t_s - N_s T_2)$$

These methods are used in getting the  
in the Test Tables, Chapter XIII., where  
called  $\Delta E$ , and  $W_R$  is the disposable work

$$W_R = (E + \Delta E).$$

The advantage over a direct calculation, is



into (425), is that the latter requires precise arithmetic, while  $\Delta E$  can be well enough found by means of the slide-rule.

An essential inaccuracy with superheat is due to the fact that the specific heat varies with the temperature; but as a fair approximation a mean value for the given range and pressure may be used in Eq. (432). And with very high superheat the adiabatic TU may drop to the pressure  $p_2$  while yet in the region of superheat, as at U, beyond the curve BM: then the constant-pressure exhaust-line will follow the curve UV, and of the total heat supplied beyond BD all under CVU will be rejected. The portion VUX can be most easily found graphically, and either be added to  $N_s T_2$ , or subtracted from  $\Delta E$  as found by (433).

As bearing on the practical importance of this last point, we may note that the curve VUT on Fig. 652 is drawn for the conditions in Test 401.3, Table 75 E, with 600 degrees of superheat, non-condensing. With about 300 degrees X will fall very near R on a horizontal line at 15 lbs. abs.; but with a condensing engine or turbine X will be far inside of R for any attainable superheat.

(b) NET PERFORMANCE OF THE TURBINE.—Of the total energy  $E$  which is available for accelerating the steam-jet, a very considerable portion is not effectively transferred to the turbine-rotor, but is wasted by being changed back to heat through current-friction, eddies, etc., and rejected in the exhaust. In Fig. 652, for instance, it is assumed that one-third of  $E$ , represented by the area GBCH, is thus lost; then the effective output is only AGHE, and the heat rejected is increased by the amount DCJK. The entropy-increment CJ is found by dividing GBCH by the absolute temperature  $T_2$ . Here we do not attempt any analysis of the losses, but simply lump them all together and show their gross effect: and it must be clearly understood that the line GH does not represent any operation in the turbine, but is used merely to divide the area in a certain proportion. Referring to Table 75 E, we see that the total thermodynamic waste is likely to lie between 35 and 45 per cent. in good turbines of fair size. But before taking up the discussion of the character and amount of these losses, we shall consider briefly what happens in a multiple-expansion turbine, where the wastes in one stage increase the supply of heat for the next.

This is illustrated in Fig. 653, for pressure-limits are 165 lbs. and 1 lb. per sq. in. The total available energy  $A_1B_1CE_4$  is divided into four parts by the lines  $A_1B_1$ ,  $A_2B_2$ ,  $A_3B_3$ ,  $A_4B_4$ . Note that the adiabatic line for the second stage each of the succeeding stages has drier steam than it would have had with simple expansion from the initial state at  $B_1$ . Of course the effect is comparatively small: thus in Fig. 653 it is taken to be 65 per cent., this being 1

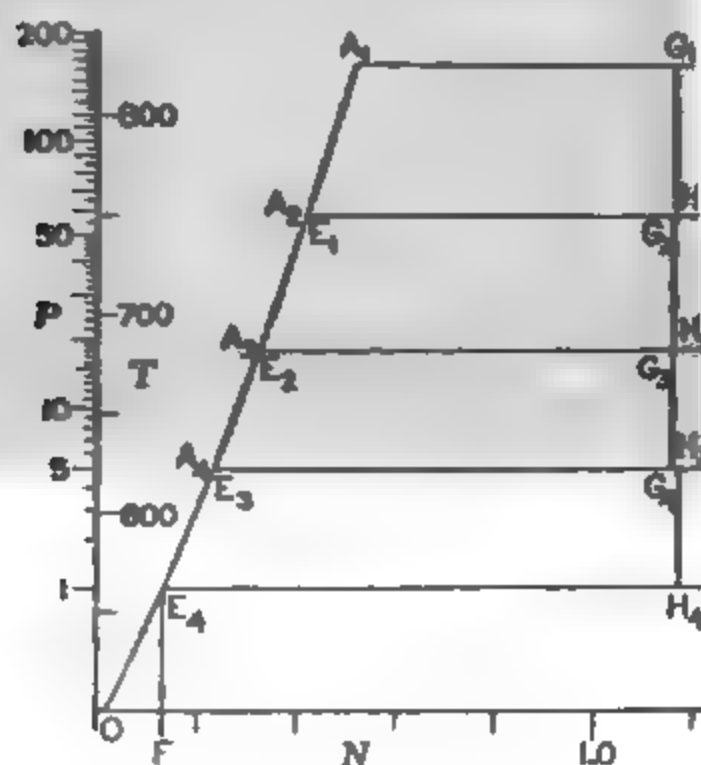


FIG. 653.—Diagram for the Multiple Expansion of Steam.

the corresponding  $ABCE$ ; while the total available energy  $A_1G_1H_1E_4$  to  $A_1B_1CE_4$  is 68.5 per cent., or 3.5 per cent. as a slight compensation for the loss of energy in the condenser.

The actual behavior of the steam in the turbine is about which, in its more intimate and exact details, is not known. The subject is much like that of the combustion within the engine-cylinder, being equally a matter of determination and of expression in terms of the results of experimentation on isolated elements of the

ess much light is thrown on the subject by such experiments, and we shall now very briefly review the methods and the results of researches which have been made.

(c) FLOW THROUGH ORIFICES.—The best group of experiments to find the rate of flow through the convergent nozzle and the plain orifice seems to be those of Professor Rateau, made in 1895–6 and first published in 1900. Selected results are given in Table 73 A, the nozzles tested are shown in Fig. 654, and the whole series is plotted in Fig. 655. The tabular quantities are reduced to a form directly comparable with the ideal jet as set forth in Tables 26 A and B. The meaning of each symbol is as follows:

- $p_1$  = initial pressure in pounds per square inch absolute;
- $R_p$  = ratio of discharge pressure  $p_2$  to  $p_1$ , or  $p_2/p_1$ ;
- $K$  = actual value of the divisor in the formula

$$W = \frac{ap_1}{K}, \quad . . . . . (435)$$

where  $a$  = area of orifice (least area of nozzle) in square inches and  $W$  is discharge in pounds per second. To put the relation into simplest shape, let  $w = W/a$  be the flow in pounds per square inch per second, so that

$$K = \frac{p_1}{w}. \quad . . . . . (436)$$

This formula was used in getting  $K$  from the original tables of results: and it is to be noted that  $K$  is numerically the same either for lbs. per sq. in. of pressure and of discharge or for kg. per sq. cm. of pressure and of discharge.

To continue with the symbols in Table 73 A,

- $K_T$  = tabular or theoretical divisor, from Tables 26 A and B;
- $K_T/K$  = coefficient of discharge, equal to  $W/W_T$ , where the latter are respectively the actual and theoretical rates of flow;
- $W_H$  = actual rate of flow in the test, expressed in pounds per hour to give an idea of the magnitude of the experiment.

The quantity plotted on Fig. 655 is the coefficient of discharge,  $W/W_T$ , on the pressure-ratio  $R_p$  as base. Curves A, B, C (for the similarly designated nozzles) are separated, each having its own scale, while N is their average to the main scale at the left, which serves for curve D also. Except at the beginning of curve D the results are very smooth and consistent, even with the irreg-

TABLE 73 A. EXAMPLES FROM RATEAU'S EXPERIMENTS ON FLOW OF STEAM.

	$p_1$	$R_p$	$K$	$K_T$	$K_T/K$	$W_H$
A	151.8	.890	108.3	103.9	.959	561
	143.7	.738	77.8	75.7	.974	707
	151.4	.466	70.1	70.5	1.003	830
	138.3	.015	69.3	70.3	1.015	765
B	77.1	.953	159.0	148.2	.933	503
	86.2	.908	115.6	110.3	.949	775
	55.7	.891	106.7	101.2	.951	542
	58.3	.474	68.0	68.7	1.011	890
	58.9	.282	67.9	68.7	1.012	900
	110.7	.019	68.8	69.9	1.016	1670
C	57.4	.984	277.	249.	.899	570
	57.6	.951	159.7	145.2	.910	992
	17.6	.864	95.7	89.8	.945	507
	22.5	.782	79.3	75.6	.956	780
	21.4	.687	70.5	69.2	.981	837
	23.2	.544	67.1	67.0	.999	951
	22.8	.432	66.4	67.0	1.009	947
	16.9	.105	65.3	66.4	1.017	712
	41.7	.058	67.6	68.1	1.022	1700
D	70.9	.965	275.6	171.1	.620	458
	46.0	.838	128.2	86.0	.672	638
	41.4	.640	92.0	68.8	.749	802
	58.8	.396	81.4	68.7	.844	1280
	54.9	.257	78.2	68.6	.877	1250
	57.5	.039	77.9	68.7	.883	1312

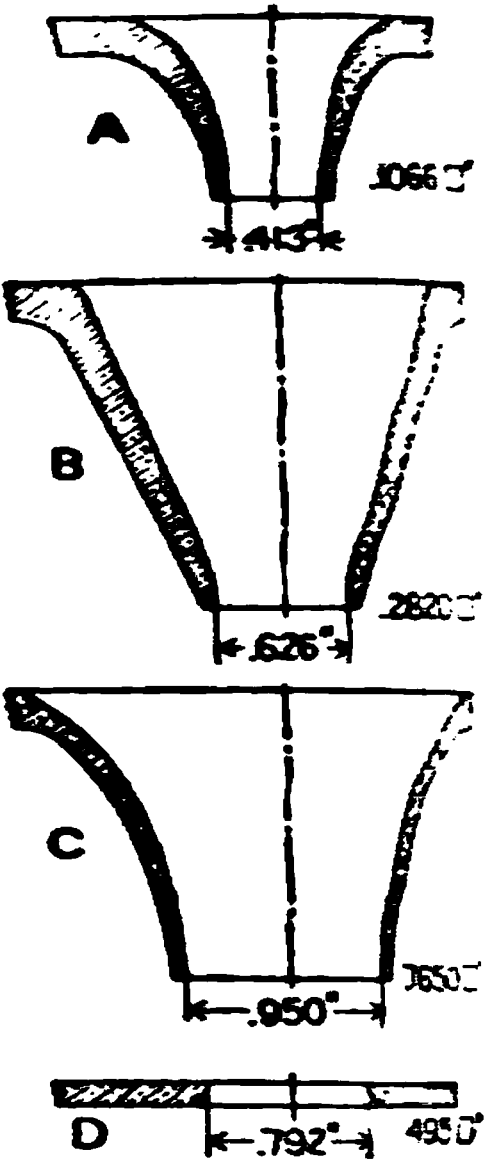


FIG. 654.—Nozzles Used in Rateau's Experiments.

larities greatly magnified by the coarseness of the vertical scale. Variation in the initial pressure  $p_1$ , as distinguished from variation in  $R_p$ , seems to have very little effect. For the converging nozzle the actual flow keeps very close to the theoretical, but the manner of change in relation, and especially the fact that the coefficient becomes greater than one with low discharge pressures, have not

been rationalized. It does not appear that the small differences in form of nozzles A, B, and C have any appreciable effect upon the flow. With the plain orifice the discharge keeps on increasing after  $p_2$  drops below the critical value  $0.58p_1$ , though at a diminishing rate.

Other experiments confirm those of Rateau, showing a similar small excess of actual over theoretical discharge with low pressure-ratios, even when a short straight tube is added beyond the con-

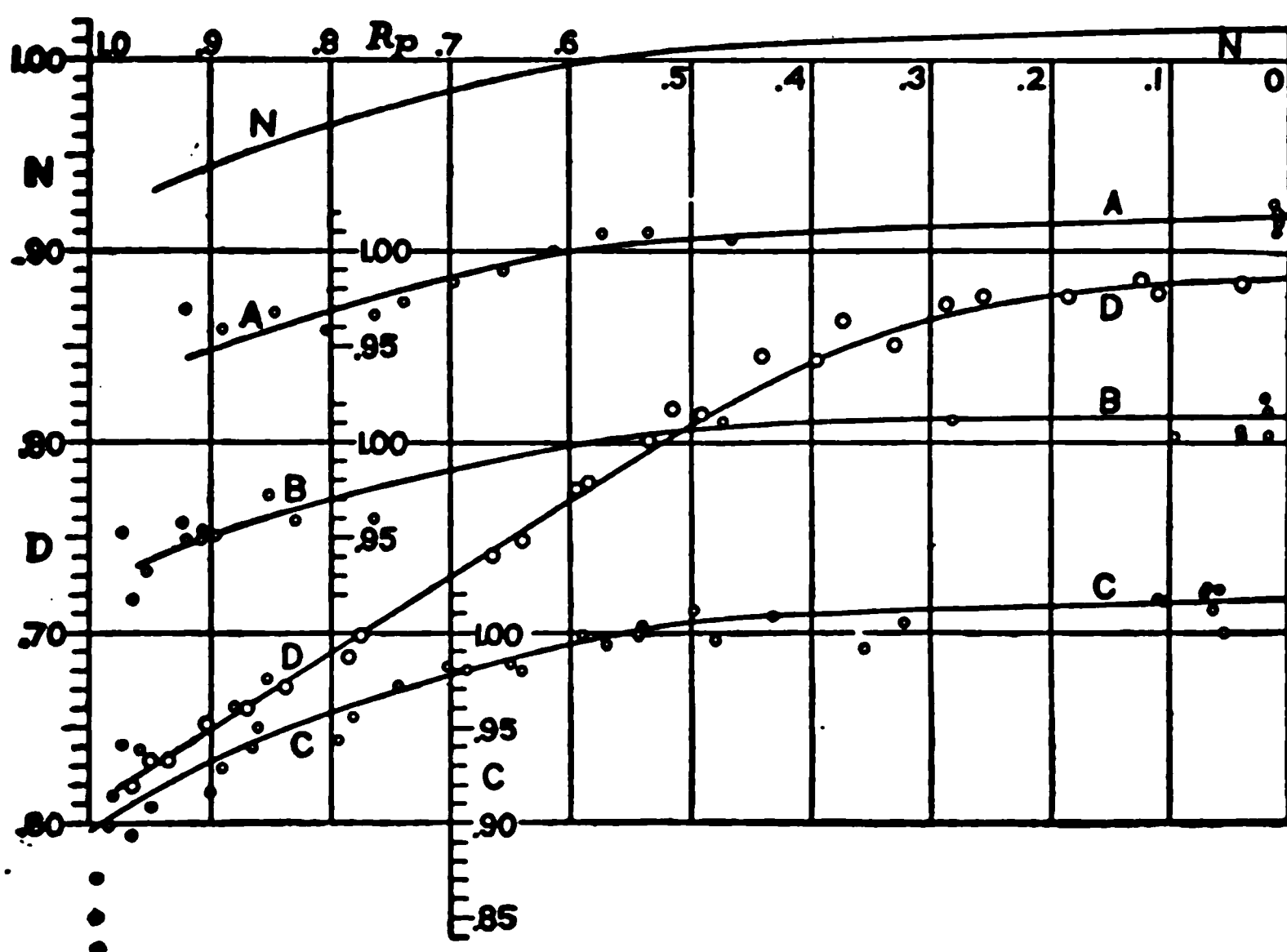


Fig. 655.—The Coefficient of Discharge as Determined by Rateau.

verging nozzle. A good collection of data along this line will be found in Thomas' *Steam Turbines*, at page 101 of the first edition.

(d) FLOW WITH A DIVERGING NOZZLE.—Some experiments have seemed to show that the addition of the diverging cone diminishes the rate of flow; but the more reliable work does not appear to confirm this view for a nozzle with proper entrance-rounding. In one of Lewicki's tests (see Art. (f)), a De Laval nozzle for atmospheric exhaust was used having the following dimensions:

Least diameter,  $d_0 = 0.238$  in.; outlet diameter,  $d_1 = 0.305$  in.; least area,  $a_0 = .0447$  sq. in.; ratio of divergence,  $a_1/a_0 = 1.52$ . Note:  $d_1$  is measured where the slanting end-surface begins to curve across the cone, or it is the largest diameter of the fully enclosed part of the cone. The total taper of the cone is about 1 in 16.

With an absolute steam-pressure of  $p_1 = 99.1$  lbs., and atmospheric discharge-pressure  $p_2 = 14.5$  lbs., making the ratio  $R_p = 0.147$ , the flow was .0664 lb. per sec.: this makes the rate  $w$  equal 0.147 lbs. per sq. in. per sec., whence  $K = p_1/w = 66.7$ . For this experiment (No. 5 in Table 1 b of the original), the steam was superheated only a few degrees: we may therefore take  $K_T$  from Table 26 B, getting 69.7, and the coefficient of discharge becomes  $K_T/K = 1.045$ . The flow was not measured directly, as this was a reaction experiment; presumably it was found by a parallel test, with condensation and weighing of the steam discharged. The excess of 4.5 per cent. seems rather large, but the essential fact is that there is an excess, just as in Rateau's experiments with converging nozzles.

In one case the diverging nozzle greatly augments the rate of flow, as pointed out by Stodola, Ed. II., English, page 68. This is when there is only a small pressure-drop, and the action is of the kind that is illustrated in Fig. 657. The diverging part of the nozzle is entirely superfluous under these conditions, so far as the proper formation of the jet is concerned; and its effect is to induce an excessive pressure-drop and a higher velocity in the throat, with subsequent rise and retardation. The velocity at the throat of the nozzle is, of course, what determines the discharge.

The information available in regard to this subject of the rate of flow seems to justify the conclusion that, with properly formed nozzles, the discharge will differ very little from that deduced from simple theory. What is meant by a properly formed nozzle will be further developed presently.

(e) VELOCITY OF THE JET.—How nearly the ideal velocity is realized at the mouth of the distributor or nozzle is a fundamental criterion of the performance of this part of the turbine, since it shows what proportion of the available energy has been effectively converted into the form which can be absorbed by the rotor.

experimental determination of this velocity is a matter of much difficulty, and some investigators appear to have followed false leads.

*The Reaction Method.*—With the first part of § 72 before us, measurement of the force of reaction offers itself as a very simple way of finding the velocity with which this reaction corresponds—using either the reaction on the containing vessel as indicated in Fig. 633 I., or the pressure on a flat plate held in the jet as in Fig. 633 II.

*Reaction of Jet-formation.*—The first idea is applied by forming the nozzle in the side of a vessel which is suspended at the end of a pendulum or of a flexible tube, steam being introduced at the point of suspension or of attachment of the tube; the opposite side of the vessel bears against a suitable dynamometer or weighing device, the scheme being outlined in Fig. 656 I.

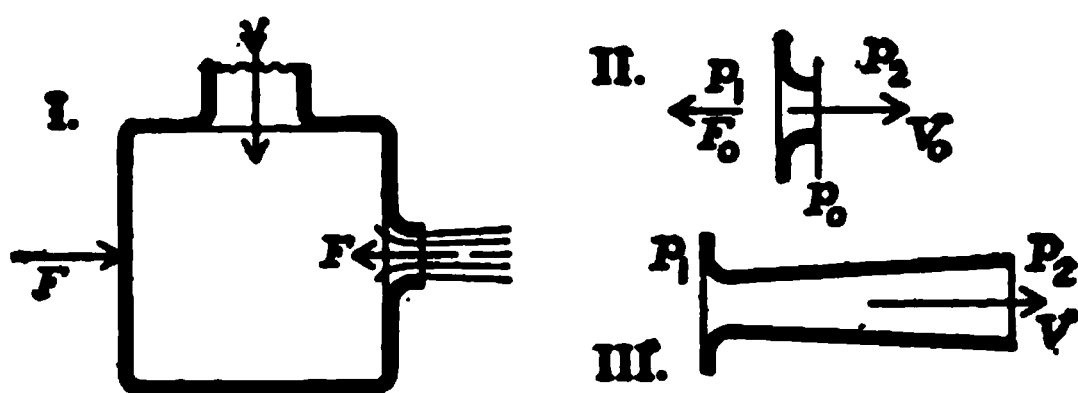


FIG. 656.—Measuring the Reaction of the Jet.

Several extensive sets of experiments have been made along this line; but a fundamental error which vitiates most of them consists in the assumption that the force measured is all dynamic reaction, due to the realized velocity of the jet. That this assumption is incorrect is shown at II. in Fig. 656. At the mouth of the plain converging nozzle will exist the velocity  $V_0$ , produced by the drop from  $p_1$  to  $p_0$  ( $=.575 p_1$ ); with this velocity the dynamic reaction  $F_0$  will tend to impel the vessel toward the left. Besides  $F_0$ , there will be a static-pressure reaction, due to the fact that upon an area  $a_0$  on one side of the vessel (the nozzle-mouth) acts the pressure  $p_0$ , while upon the corresponding area on the other side acts only  $p_2$ . In general, if  $p$  is the pressure and  $a$  the area at the mouth of the nozzle, the jet having the velocity  $V$ , to which corresponds the

reaction  $P$ , the total force tending to move the vessel  $T$  is

$$P = F - p - p_2 A \quad \dots \dots \dots (1)$$

Only when the pressure in the jet at the mouth of the nozzle is equal to  $p_2$  or when the whole operation of acceleration and velocity at  $p_2$  is carried out within the nozzle does the total force agree with the dynamic reaction—the case being illustrated by Fig. 656 III. It appears, however, that when the drop in pressure is too much for the drop from  $p_1$  to  $p_2$  so that the pressure in the jet falls below  $p_2$  and then rises to this value at the nozzle exit, the action being all carried out within a nozzle which is part of a movable vessel, that then the net effect is a dynamic reaction due to the velocity at the mouth of the nozzle. The latter force due to the highest velocity is partly balanced by the effect of the retardation in the nozzle towards its mouth.

An analysis of the exact manner in which the reaction force acts upon the containing vessel, involving a study of just how the pressure varies and the current is accelerated, might be of interest but it is of no practical importance, in view of the fact that we pass over all these details and go right to the resultant effect already pointed out in § 72 (a).

Experiments by this method are valueless without a measurement of the pressure in the jet at the mouth of the nozzle—a determination not included in any that have so far been made and published.

(f) REACTION OF JET-DESTRUCTION.—This heading describes what takes place when a flat plate is set squarely in the path of a jet, as in Fig. 633 II., so as completely to destroy the forward movement of the current, deflecting it all out sidewise. To illustrate what can be done by its method, we will consider briefly the experiments of E. Lewicki at Dresden, published in *Zeitschrift des Vereins deutscher Ingenieure*, 1903, page 491.

The nozzles used belonged to a 30-H.P. De Laval turbine (Test 401, Table 75 E); one has already been described in Art. 10, the other was a converging nozzle with a 10° taper—in effect first reversed. The steam-pressure was 100 lbs. per sq. in.



ute, and the plate on which the jet impinged was held at distances ranging from 0.2 in. to 9 ins. from the end of the nozzle—the latter being some distance from the mouth proper, on account of the slant at which the nozzle is cut-off. The dynamic pressure of the jet being measured, the velocity was calculated from it and compared with the ideal velocity that would be produced by a perfect transformation of energy with adiabatic expansion. The results secured were as follows:

Using the converging nozzle, the velocity-loss varied from 0.128 when the plate was close to the nozzle-end to .08 at 2 ins. distance, .05 at 3 ins., .033 at 4 ins., .043 at 6 ins.; each of these fractions showing the difference between actual and ideal velocity, expressed as a part of the latter. These tests were all made with about 100° F. of superheat. With the diverging nozzle, properly proportioned for the pressure-drop, the loss varied from .05 or .06 at 2 ins. to .035 at 6 ins. In this group the superheat was varied from zero to something over 100°, and for the distance 6 ins. the velocity-loss ranged from .043 to .033, decreasing as the superheat was greater. As might be expected, the results show irregular fluctuations, yet they are on the whole quite fairly consistent.

In interpreting these results, we note first of all that the deficiency in kinetic energy is about twice as great as that in velocity; thus if the actual velocity is 0.95 of the ideal, the actual energy will be  $0.95^2$ , or 0.9025 of the total available energy. It appears, then, that with the diverging nozzle the jet is from 10 to 12 per cent. short on energy near the nozzle-end, while with the converging nozzle the shortage appears to be as much as 24 per cent. The meaning of the smaller values got when the plate is farther from the nozzle is that the jet picks up air from the surrounding atmosphere, with no loss, but rather a gain at first, in the momentum of the increased moving mass, even though there be a loss of kinetic energy. But these distant measurements are obviously of no value as showing the efficiency of the nozzle in producing a jet applicable to the turbine-wheel.

(g) MEASURING THE PRESSURE IN THE JET.—A method of experimentation which leads to many interesting and some highly useful results is that of the exploring tube with a small hole drilled

at one point in its length, which is local, and can be moved endwise so as to place the steam-jet at various points—the scheme of Fig. 28 (e), besides being outlined in Fig. 63. A large number of experiments have been made, line, by Stodola and others, throwing light on the phenomena of jet-formation. A brief statement of what can be drawn from these data is given in the following articles.

(h) ACOUSTIC VIBRATIONS.—When a short converging or parallel nozzle, the length anywhere from 0.7 to 0.1, there is easily set up acoustic vibrations. This type of vibration is that is, backward and forward in the direction of flow, or here along the steam-current; it consists of establishment of alternating zones of higher and lower pressures, fixed in location, and giving the pressure-curve a wave-like form, which absorbs some energy, diminishing that available for the jet form.

If a diverging nozzle is so proportioned that its mouth is reached by a continuous drop of pressure from discharge-pressure  $p_2$ , then the current flows smoothly and evenly. But if the nozzle is such that the jet flows into an atmosphere of pressure  $p_0$  and the nozzle expands too much, so that the pressure at the mouth is of the character typified in Fig. 657, strong vibrations are set up.

There is here a contradiction, not readily apparent at hand (Stodola's curves): with a converging nozzle, with a coefficient of expansion 0.58, the jet flowing into its own pressure  $p_0$  but with a diverging nozzle fitted to any given value of  $p_2$ , there is little or no vibration. The pressure at  $p_0$  (at the choke of the jet) is the same in both cases. In the particular medium may have some effect on the question for the physicist. The subject of acoustic vibrations is not fully mapped out, and we are no

conclusion that would make a stage with small pressure-drop less efficient, in this particular respect, than one with large drop. The important practical result is the strong testimony to the effectiveness of a properly designed diverging nozzle.

(i) **DIFFUSER ACTION.**—While the nozzle with a proper ratio of divergence delivers the jet of maximum efficiency, it is most essential that the nozzle be not carried too far, or expanded too widely, so as to set up the action represented in Fig. 657 and plotted from experiment in Fig. 64 (from Stodola). If the atmosphere on the discharge side is kept away from the jet and the latter flows forward within an enlarging channel, it will drop well below  $p_2$ , with the development of a correspondingly high velocity; but after the minimum pressure has been reached (depending on conditions not yet made determinate)

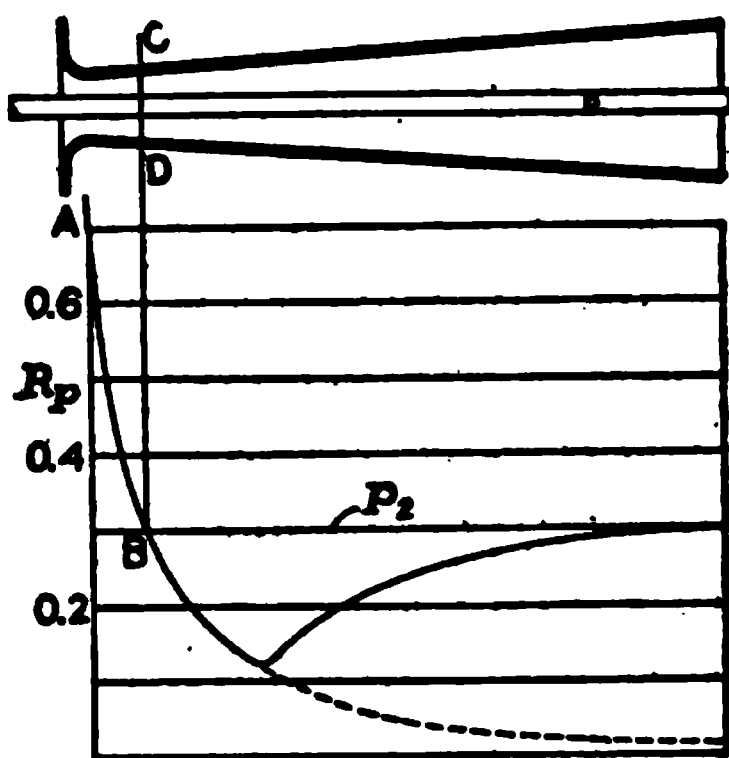


FIG. 657.—The Nozzle with too much Expansion.

there is a gradual rise of pressure, with retardation of the jet; and the final velocity, at the mouth of the long nozzle, is far less than that which would have been secured if the latter had been cut off at the point where  $p_2$  was first reached, at CD in Fig. 657. Over-expansion of the nozzle is most decidedly to be avoided in the design of the turbine.

(j) **ENERGY-LOSS IN THE NOZZLE.**—An indirect method of getting at the real velocity of the jet, less obvious than those discussed in (e) and (f) but rather more promising of reliable results, is based upon pressure-measurements along the nozzle. Having found by trial the exact rate of discharge from a certain nozzle, we can calculate how the pressure ought to vary along its axis, first with ideal adiabatic expansion, second under the assumption that the energy actually put into the jet is less than the ideal amount by a certain proportion. From these calculations, which give directly the cross-section at each pressure as in Fig. 46 or Fig. 48,

we can draw, on the nozzle-length as manner of pressure-drop for zero, ten, energy-deficiency. By plotting the observed curves, we can see just what is the energy-loss, and thence easily pass to its velocity.

Stodola gives an example by this method. He used the nozzle shown on Fig. 64, 1.5 inches in diameter by 6.3 ins. long, with steam heated about 30° F. At 11.5 lbs. abs. pressure from the outlet, the observed pressure loss was 18 per cent. of energy-loss; at the outlet, where the pressure indicated a loss of about 18 per cent. of energy-loss. Lewicki's tests are fairly consistent with this. He concludes that for nozzles from 0.2 to 6 inches (usual in De Laval practice) the energy-loss is from 10 to 15 per cent. In large multiple-stage turbines, depending a good deal upon the steam pressure, Friction is, of course, the chief source of energy-loss.

The weak point in the scheme just described is of measuring the pressure with the device shown. Also, it requires a forbidding amount of calculation; however, be greatly facilitated by graphic methods.

(k) DEFLECTION OF THE JET. -Beyor has been done in the way of effective experiments of the steam-current; and it does not appear that a large part of the subject is open to detailed examination. There is room, however, for some examination of the action of a steam-current in a curved path to be carried out by a device analogous to that shown. Nothing along this line has as yet been published.

In Fig. 634 I. the current is represented by a number of stream-lines or narrow ribbons of fluid. If the particles move smoothly around the curve, the particles will follow the curve. The cross-section of the current at entrance and exit, and the only acceleration is transverse or centripetal. It is easy to see that this can be even approximately realized only when the



its width only a small fraction of the radius of the curve; but the development of a rational and consistent idea of what takes place in a broad channel, like those between the blades in Fig. 616 or instance, is a very difficult problem.

If we could adhere to the steam-line hypothesis, the action might be somewhat as represented in Fig. 658 I., where the outer streams are crowded together on account of the centrifugal force of the whole current, which has been discussed in § 72 (b). A number of knotty questions now arise, as follows:

The outer part of the stream being compressed, what is the linear effect (along the current) of this increased pressure or stress within the steam-substance?

Will the outer and inner streams traverse their paths of different length in the same time? If so, how will the differences in velocity be produced? If not, will the streams slide on each other, so to speak?

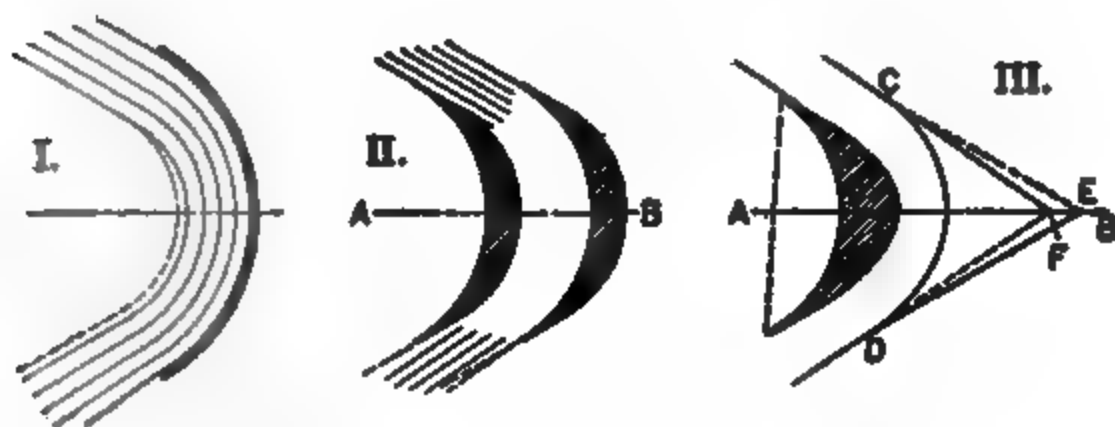


FIG. 658.—Behavior of the Jet in the Bucket or Vane-channel.

In what manner will uniformity of pressure throughout the current be established at exit?

Without trying to answer these, it is evident that the idea of "parallel" ribbons will have to be abandoned; then considering the stream as a whole, with the first part of the curved guide-surface cutting obliquely across the line of flow, we see that this surface will not only deflect the current (accelerate it transversely) but will also exert a component against the entrance velocity, producing a linear retardation. Instead of sweeping smoothly around the curve, the jet will pile into the bucket in a confused and tu-

multuous fashion, with a rise in the steam pressure in the confined space; and then from this point the escaping jet will be again formed in the direction that which prevails in the reaction turbine.

As to the direction of the outflowing steam, it is tangent to the outer edge of the bucket at the point where it is tangent to the curve at this point, but it is at an angle with the center-line AB. About the same angle as assumed from theoretical considerations in the case of the stream-directions at entrance and exit. In this connection refer back to the discussion of the reaction turbine.

The influence of the considerations just mentioned on the later design of buckets for the Curtis turbine is shown in Fig. 658 III. The converging of the steam against the corner of the bucket at the narrowest point there is a channel of discharge. The centrifugal effect will grow less toward the outer edge of the vane-angle on this side compensating for the jet to swing outward. Further, the bucket is covered by shrouding the blades, or covering the higher pressure produced by the piston bucket and keep it in line for the production of the reaction.

Referring to Fig. 623 we see much more channels than in the single-impulse turbine. In Fig. 619, the need for maintaining a fairly perfect balance of the steam pressure is seen in the exceedingly broad gaps between the buckets. With narrower channels, there will be more of the relatively greater surface over which the steam can flow.

(1) STEAM FRICTION.—The idea that the losses of mechanical energy on account of the current on confining surfaces might be estimated by the velocity and pressure, and velocity if only the coefficient of friction is known, following the line of the discussion, in § 619. Such a calculation would be greatly complicated by the variations in pressure and velocity over the surface; but the great obstacle is the lack of

between pressure and velocity on one hand and frictional resistance on the other. It is understood that the friction itself (force, not coefficient) varies almost directly as the pressure (or inversely as the density) and as the square or some higher power of the velocity; but the numerical relations remain to be determined, and it does not appear that their determination is at all an easy problem. An important and apparently well-established fact is the marked influence of superheat in diminishing steam-friction.

(m) ROTATION LOSSES.—Besides the friction of steam-current on metal, which absorbs a portion of the kinetic energy of the former before it has a chance to be effectively taken up by the turbine blades or buckets, there is another friction-action through which some of the energy of the rotor is wasted, namely, the friction of the rotor-surfaces upon the “dead” steam which, as an atmosphere, fills all the spaces outside of the path of the main working-current. This dead steam is not quiet, but is in more or less violent churning or whirling movements; and it is through this useless motion that the lost energy is changed back to heat and carried off in the exhaust.

The scheme for finding the amount of this loss is to revolve the rotor, or a part of the rotor, in a steam atmosphere like that which surrounds it in normal working. This is likely to give resistances a little too great, for the blades develop relatively much more “friction” than the disk-surfaces; and when the turbine is working some part of the blade-ring is always in the steam-belt, where the active current moves with the blades and churning action does not exist. A sample of the results got by this line of experiment is given in Fig. 659 (compare Fig. 692), where the steam-pressure about the wheel of a De Laval turbine is the base, and the effective power required to drive the wheel and gearing—motor input less electrical losses—is the abscissa. Lewicki's tests, already referred to, gave similar results.

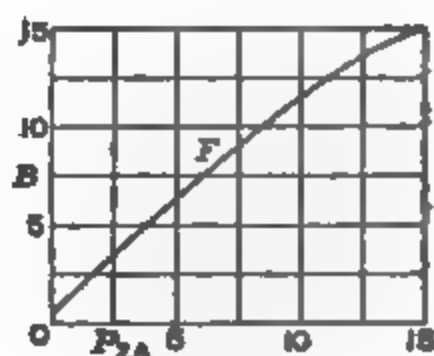


FIG. 659.—Total Rotation-losses in a 50-H.P. De Laval Turbine: see Test 402, Table 75 E.  $B$  = brake horse-power.

... disks for turbines ... by H. H. ... 1957.

... of the turbine is ... to be included, ...

... more than one ... here to the ... of the cellular type ... fitting closely ... see Figs. 620, 621 ... by Fig. 622 ... the ends of the ... because the full ... it is evident ... yet do work at the ... and the blades ... very very small.

... a mechanical problem, depend- ... of rotor and ... will be avoided, and ... construction. The ... balance-pistons, ...

... is practically impossible to ... but it is obvious that ... accounting for quite a ... actual performance.

... By this is meant the width of the ... of nozzles or distributors and the ... with the Riedler-Stumpf type of ... The reason why small clearances ... in impulse turbines, besides the liability ... about the working current, is shown ... If the stream flows around and along a cylinder ... a helical channel, it will have a certain width; if the guiding ... still with the same circumferential distance between



ddenly made parallel to the axis of the cylinder there will be a abrupt and decided increase in width. In other words, the current from an inclined nozzle or group of nozzles will tend to swing into a direction parallel to the axis, slowing down to fill the wider cross-section thus accommodated; and the need of preventing such action is one reason for running the blades close to the nozzle-mouths. In the reaction turbine, and on the discharge side of impulse blades, the line of flow is so nearly axial that width of clearance is of little or no importance.

(p) ANALYSIS OF TOTAL ENERGY-LOSS.—This brief presentation, contained in Arts. (c) to (o), is truly representative of the general state of knowledge of steam-action within the turbine, in that it gives much more information about the initial operation of forming the jet than about the behavior of the jet after it gets to work. The total energy-loss can be easily enough determined, and the data in Table 75 E show that the deficiency in actual output is somewhere near the same as in the engine, having a full range from 30 to 60 per cent. and a usual range, for ordinary good practice, from 35 to 45 per cent. The work of quantitatively analyzing this loss has been greatly hampered by the lack of some means of measuring the condition, or quality, or heat-content of the steam from point to point of its expansion: almost the only expedient available has been the measurement of pressure and a rough inference therefrom, after the manner set forth in Art. (j). Improvements in steam-calorimetry, especially the electric calorimeter perfected by Professor Thomas, which can accurately measure the quality of steam at any pressure, now give promise of a method which shall serve the turbine as effectively as the indicator has served the engine.

#### § 74. Design and Construction of the Turbine.

(a) CROSS-AREA OF THE STEAM-CHANNEL.—In laying out an engine which shall develop a certain power, the principal step is to determine proper sizes for the cylinders, and the machine is then designed around these volumes. The analogous fundamental

determination in the design of a turbine (through nozzles and buckets) along which the steam will flow. The essential requirements are that the buckets be the right size to carry the necessary amount of steam and be properly graded in sectional area so as to insure correct expansion of the steam.

From the known limiting pressures and the work per pound of steam can be found, a reasonable value—say 60 per cent.—dynamic efficiency \* being assumed, the initial pressure of the steam becomes known and leads at once to the initial velocity required. The initial nozzles must pass the steam at this velocity due to the pressure-drop in the first stage. The successive critical areas must be proportioned with reference to the initial area of steam-expansion furnishing the needed work.

The simplest case is the one-stage turbine (of the Laval type), with but one nozzle, or gear, which can be laid out with sufficient correctness by the use of the theoretical computations in §§ 24 to 26. The initial area of ideal steam-jet is determined.

Next in order comes the many-stage turbine, taken first with uniform vane-velocity, and then with diameter in all the stages. With stage velocity constant the steam must reach the same velocity in all the stages; then the areas of the successive cross-sections must be as does the specific volume of the steam. This is illustrated in Fig. 660, where the buckets are shown with  $B_1, B_2, \dots, C_4$  on Fig. 653, each section representing an adiabatic expansion in the nozzles of a stage. The area is determined by the total supply of energy, and the steam supply including the waste in the preceding stage. If the discharge-velocities are equal, it is very evident that the areas must be exactly proportional to the specific volumes.

The statement just made holds equally for the

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\* See § 75 (d) for precise definition.

impulse turbine with disks of uniform diameter (Curtis type), so long as we consider only the initial action in each stage. But when there are vane-rings of differing diameter and velocity, so that the later stages must transform larger quantities of energy—see § 72 (j)—the relation of channel-area to specific volume will change with the vane-speed. That is, as the steam-velocity is increased the area needed for flow becomes relatively smaller.

These laws govern the reaction turbine likewise, both as to the relation between area and specific volume and as to the effect of changes in diameter and velocity.

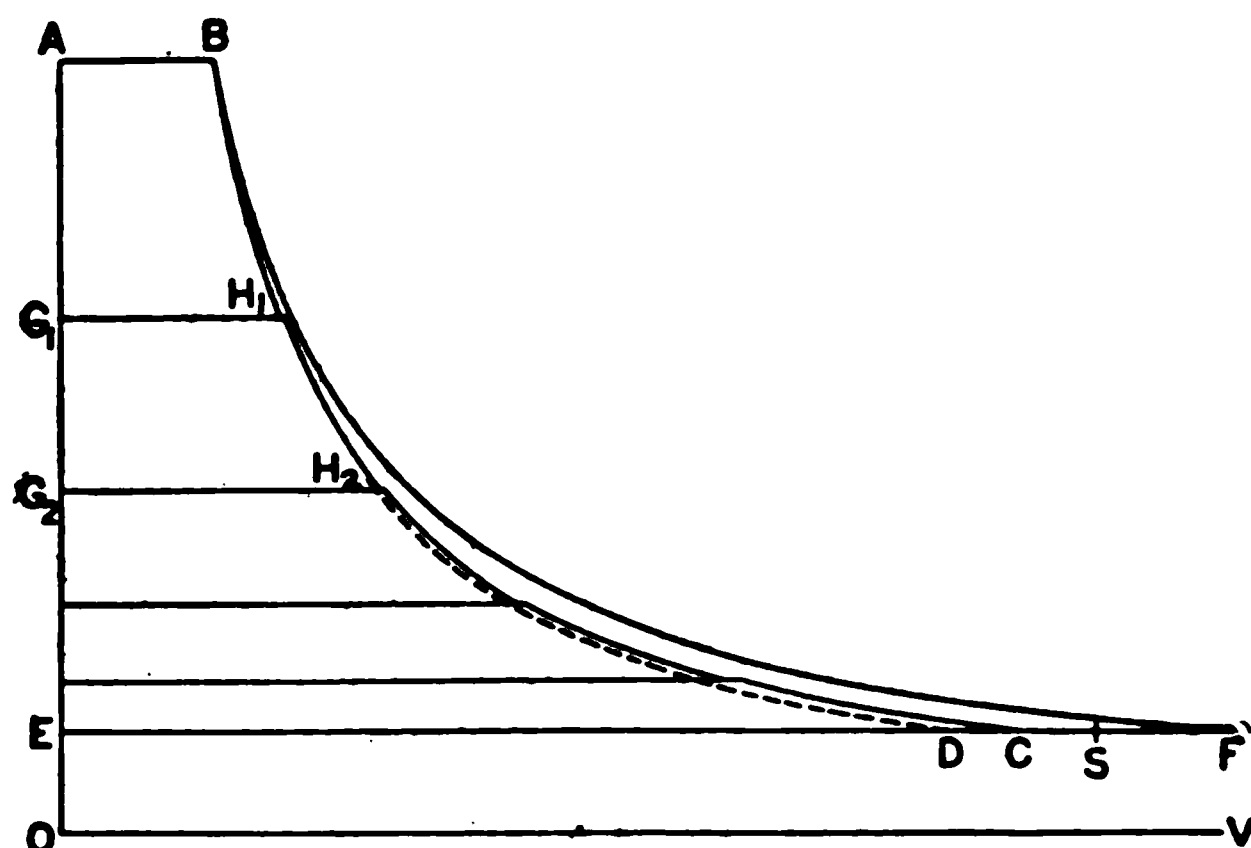


FIG. 660.—Pressure-volume Diagram for a Multiple-stage Turbine—compare Fig. 653.

In this elementary discussion we have passed over the secondary actions which are likely to modify the simple relations just set forth, though not to any very considerable degree. Thus on account of greater losses by leakage and friction at high pressure, it may be desirable to give the higher stages a little more than their true proportion of ideally available energy—in other words, to use higher jet-velocity in the upper stages, in comparison with the vane-velocity, than in the lower stages. In the Parsons turbine, on the other hand, it is usual to have nearly or quite the same effective area through all the vane-rings of a given size and depth (see Fig. 629); then within the group the manner of pressure-drop

will be such that the earlier stages will have less loss than those toward the end.

The calculations for a diagram like Fig. 622 are complicated, becoming quite laborious with many stages. They can be greatly simplified, however, by using the entropy-temperature system. For the purpose of the more advanced works dealing with the steam-turbine; but we have here added, as a check on the relations, the curve of constant steam velocity, which would take place if, in the free expansion, there were no effective abstraction of mechanical energy from the jet is used up against friction. This curve of zero-efficiency is one limit of the actual curve of steam-conditions. BD, which implies the full abstraction of energy, and the actual curve of steam-conditions. It is to be noted that BF runs out into the saturation-curve from B coming to S. S has already been described in § 28 (b). It is just remarked, all these matters can be represented and discussed by the entropy-temperature-pressure-volume method.

Some discussion of the action of the supply of steam and thus the manner of operation of the turbine will be found in Art. (k).

#### (b) CHANNEL-VARIATION WITHIN THE TURBINE

—What we have so far considered is a question of the flow through a nozzle, or a question of the flow through a single stage. To get a clear idea of what happens when the steam passes through several rows of moving vanes, or through the nozzles, is a more difficult matter. It is pointed out in § 72 (i) that if the steam-velocity is small in comparison with the vane-velocity, the combination of the vane-velocity with it—i.e., the resultant velocity of § 72—the change in obliquity of vanes at entrance and exit will give the resultant velocity. Compare Fig. 623. But because of the loss of energy on account of friction, shocks, and eddies

sary to use increasing radial depths, as shown in Fig. 621. If this is not done, the current will tend to "back up" in the channel, so that there will be a dropping pressure from nozzle to final exit, except as such an action is obviated through the sidewise diffusion of the current which may easily take place when there is partial peripheral admission; but any great amount of this spreading out into the dead spaces is bound to cause excessive eddy-losses.

(c) THE ROTOR of the turbine must have the proper form to carry the blades and the needed strength to resist the forces which come upon it, and must be as nearly as possible in centrifugal balance, so that its rapid spinning will not cause vibrations. Typical forms of rotor have been illustrated in § 71, and the matter of blade-fastening will be taken up presently. As to strength, the major stress is generally due to centrifugal force, although resistance to flexure by weight may become of first-rate importance in a long horizontal turbine. The tangential driving forces, due to the steam, are relatively insignificant.

High-speed turbines have rotors of disk form, which resist centrifugal force in a manner quite different from that of the thin ring, which has been discussed under fly-wheels, in § 36 (i). Without going into the complicated mathematics of the subject, we can get an idea of the conditions in a disk by considering Fig. 661. Suppose that a triangular element is cut loose along OA and OB, and moved outward in the direction OH, then all points on the radial edges OA and OB will have the same displacement from their original positions, but the displacement at C, for instance, will bear a much higher ratio to the arc CD than will that at A to the arc AB. With this simple geometrical illustration of the fact that a certain radial distortion (absolute, not relative) will have a much greater effect near the center than near the rim, we can easily see the reason for the following well-established facts:

Radial tension plays a very large part in resisting centrifugal force, acting to help the circumferential or ring tension.

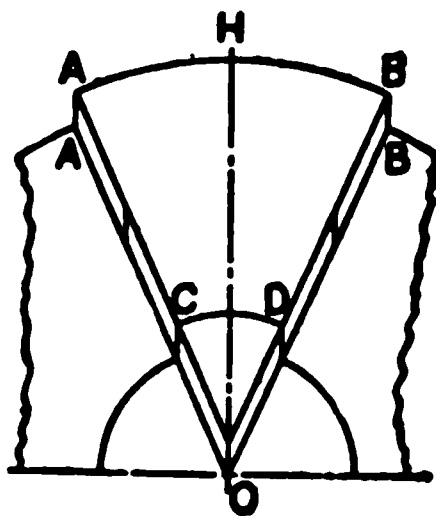


FIG. 661.—Diagram to Illustrate Stress in Disk.

The inner ring-elements of the disk are subjected to a large centrifugal force; but the tendency of the elements to



FIG. 662 Sections of De Laval Disks. I. Rotor of Larger Turbines; II. Detail of Shaft-fastening in Fig. 617.

move bodily outwards, the higher stresses towards the rim. In other words, an elastic deformation corresponding to the differential stress with elasticity, produces a stretch towards the center. For uniformity of radial stretch, the disk must be of uniform thickness.

To get uniform thickness, the disk must be thicker near the rim. This needs the machine to be of uniform thickness.

If the disk has a higher stress at the rim, the rim must be thicker.

These principles

of design in the De Laval turbine are: I. hub on the small end of the shaft; II. partly to compensate for the hole. Large form sketched in I.

The drum-rotor of the Parsons type is subjected to a much lower centrifugal force than the hollow rotors can be more accurately balanced by construction than when cast in one piece.

(d) BALANCING THE ROTOR.—The first step is to bring the center of mass to the rotation-axis is first to get perfect geometrical symmetry that is attained by the appliances, and then of trying for and correcting eccentricity by special experiment. To overcome the defect of absolute perfection, and to avoid the expensive work in balancing, De Laval devised a method of which the action depends upon the following principle.

If an eccentric mass is attached to a rotating body, the whole mass be thought of as concentrated

then the centrifugal force of this "particle" will act radially outward, tending to deflect the shaft in its own direction, and producing a pressure of shaft on bearings which constantly changes in absolute direction. This condition has been extensively discussed under Shaking-force and Counterbalance in § 40.

But when we consider the mass in its actual distributed state, there comes in another action which reaches considerable magnitude only at very high speeds. A spinning disk tends to get into the simple, stable state of rotation about its own center of mass—this tendency being closely analogous to that of the same disk to preserve one fixed plane of rotation, or to what is called gyroscope-action.

In the case of any nearly balanced rotating body, as the speed increases the eccentricity effect at first increases very rapidly, to a maximum at what is called the critical speed; then the other tendency predominates, and if the shaft is flexible or the bearings a little loose, the body will spin quietly on its true axis of symmetry.

At the speeds usual in large multiple-stage turbines, the degree of precision reached with good machine-shop work gives a sufficiently accurate balancing for practical freedom from vibration.

(e) BEARINGS FOR TURBINES.—In the characteristics of high speed but of a generally uniform load to be supported, the turbine is under conditions which belong rather to the electric generator than to the steam-engine, and the design of this part follows the lines of the former machine. At very high velocity, true alignment and geometrical perfection of the rubbing surfaces are of the first importance, as insuring a uniform distribution of pressure; and with these there must go ample lubrication, supplied by an oil-pump system. In turbines of any size the bearing-shell proper is solidly held in the framework of the machine, but has usually a spherical seating to permit self-alignment with the axis-direction. The bearing on the inner side of the De Laval wheel—see Figs. 617 and 664 I.—is free to move sidewise also; but a closer examination shows that this is not a bearing at all, in the sense of supporting the shaft, but serves rather as a stuffing-box. The pivot-bearing of the Curtis turbine is in many respects a special case, but it has been fully enough described in § 71 (g).

The design illustrated in Fig. 663 represents the smaller and quicker where some flexibility is secured by between the inner and outer shells of tightly fitted, and the oil-films which cushioning effect. The collar-bearing wise motion, with the particular purpose to prevent rubbing in the groove or labyrinth. The upper and lower halves are adjusted in opposite directions by the lines S, S. The worm at the right

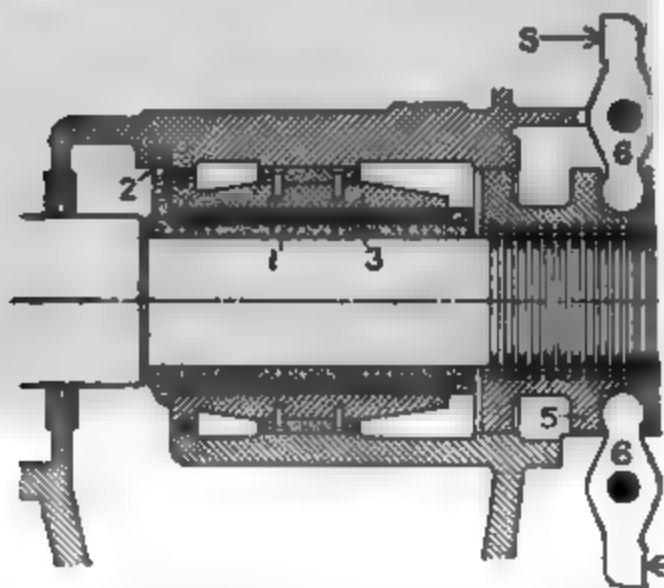


FIG. 663.—Bearing at Governor End of Westinghouse Turbine.  
Right-hand End of Fig. 662.

which carries a bevel-gear for the vertical motion of the admission-valve mechanism, and a crank to drive the oil-pump.

(f) STUFFING-BOXES.—This name, derived from the fact of implying close contact and pressure of the packing on the shaft or spindle, is applicable to the corresponding parts of the turbine as describing the function of preventing leakage in the manner of performing that function. At the present time, a few cases; but the more generally accepted idea of avoiding the contact of rapidly moving parts, allowing but a very narrow (and frequen



sage for the flow of steam. A number of special devices and methods are used, as will be made clear by the examples collected in Fig. 664.

Plain contact-rings are seen in Fig. 621; these rings are made of carbon, are held in place by brass skeleton-frames, are pressed inward by light helical tension springs, and bear upon a brass bushing on the spindle. Referring to Fig. 653, which shows typical conditions for a Curtis turbine, we see that this packing will have to retain a pressure of something like 45 lbs. above atmosphere; and with the multiple effect of a series of rings, leakage can be prevented with only a comparatively small contact-pressure. Another example of simple contact is given in Fig. 664 I., where air must be kept out when the turbine is running on vacuum. A passage can be opened only by the forcing out of the film of oil between journal and bearing, which would require more than a pressure of 12 or 13 lbs. per sq. in. There will evidently be, however, some tendency for the oil to work gradually into the steam-space.

The essential feature of the scheme shown at II. and III. in Fig. 664 is the annular space  $S_1$ , kept filled with steam of nearly atmospheric pressure by means of a reducing-valve from the steam-supply and a relief-valve to the condenser. There will be a gradual flow through the very narrow passage permitted by the sleeve at B, outward to  $S_1$  at the high-pressure end, inward from  $S_1$  at the low-pressure end of the turbine—all the “stuffing-boxes” being piped to form a common system. In effect, the turbine is surrounded by an atmosphere of steam at the points where leakage can occur; and then this steam-atmosphere is shut off from the air by lightly fitting segmental contact-rings, tied together by helical band-springs and pressed inward axially by compression springs which exert the force marked  $F$ . Oil is supplied at L. The design at III. adds labyrinth packing, both in the stuffing-boxes and between the wheel-chambers—compare Fig. 620.

A different idea is shown in Fig. 664 IV., which is used to prevent the leakage of air into a steam-vacuum. There is, of course, a fairly close running fit between the bushing B and the rings C, C; but the passage is closed or sealed by water in the annular space

about the collar D: this water has a dynamic pressure, acting radially outward, and produced by little vanes on the sides of it which are formed like those on the "impeller" of a centrifugal pump. This arrangement has the further advantage that by circulating the water the stuffing-box is kept cool.

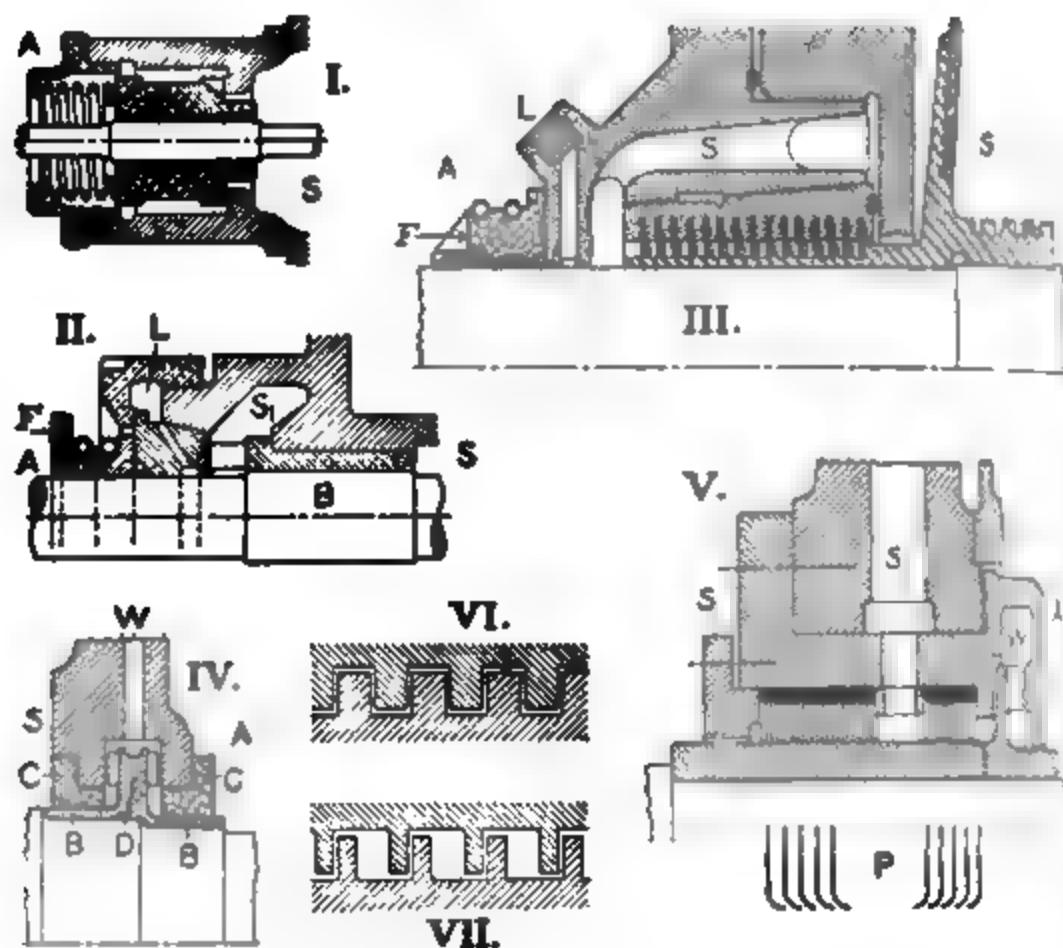


FIG. 664.—Packing Devices for Turbines: I. De Laval, from Fig. 617; II. Rateau, as in Fig. 620; III. Oerlikon-Rateau, late design; IV. Parson, from Fig. 629; V. Sulzer, Fig. 630; VI., VII. Two types of Labyrinth Packing.

In V. the last two ideas are combined, together with a special form of labyrinth packing, made of thin brass plates. These plates or washers, only about .005" thick and separated by narrow rings of thicker plate, are bent or flanged where they touch the shaft as shown by the enlarged detail below the main sectional view.

*Labyrinth Packing.*—This device, having its most important application in the prevention of internal leakage, takes the two typical forms shown at VI. and VII. in Fig. 664. The first mere

provides a tortuous passage for the steam, of nearly uniform cross-sectional area; the second and more effective form adds the further retarding influence of abrupt changes in the area of the channel. The "comb" type in V. has the advantage that it fits a plain cylindrical bushing on the shaft, but looks as if it might degenerate, in service, into something not much better than the plain sleeve in II.

(g) NOZZLES AND DISTRIBUTORS.—As regards the construction of these parts, one typical method is represented by the De Laval nozzle, which has the form given it by the operations of turning, drilling, and reaming.



FIG. 665.—De Laval Nozzle, with Shut-off Valve.

It is held in place by friction (plus the steam-pressure) and can be pulled out by means of a withdrawing tool that screws over the inner end. The Riedler-Stumpf nozzles are likewise made singly, but after machining are pressed to a rectangular shape toward the mouth, and may also be bent to fit certain holding devices, as in Fig. 624 V.

The high-pressure nozzles of the Curtis turbine, with the profile in Fig. 623, are formed in the casting by means of cores, then filed

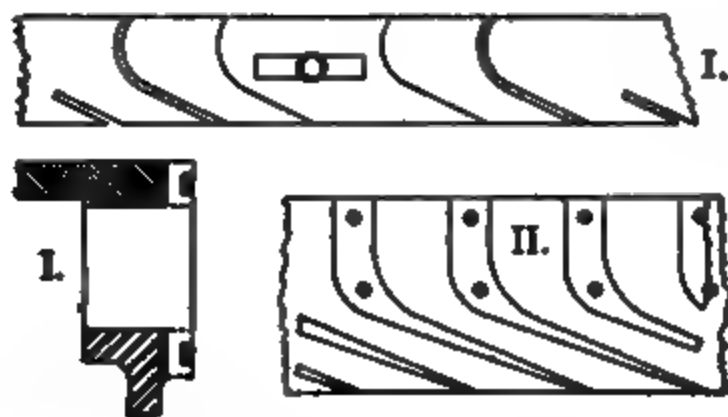


FIG. 666.—Built-up Nozzles. I. Zoelly Design, Fig. 619; II. Type of Sulzer, Union, Elektra, etc.

out smooth; they are round in the throat, changing to a rectangular section toward the mouth. The low-pressure distributors, shaped like those are made of pieces of ordinary soft

sheet-steel, which are set in the cores as diaphragms. For the last stage in Fig. 62 circumference, and the cross-struts are plates.

A more expensive but more precise and efficient distributors of the curved-plate type is one in which inclined slots are milled (sawed) in the rim of the cover-ring, and the steam plate has projections which go into these slots. On the cross-struts are nozzles. The steam pressure-difference on the diaphragms is used for the construction of nozzles for large pressure-drop stages. Partition-pieces are fastened between the cover and the axial-flow turbine, between plane surface and radial.

(h) VANES OR BLADES.—There is a wide variety of these essential working parts, and a few typical designs may be given.

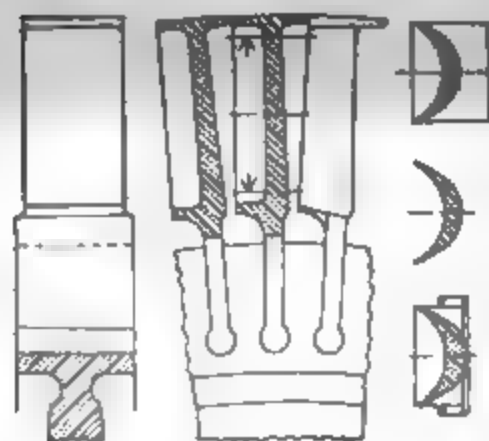


FIG. 667.—De Laval Turbine-blade.

Fig. 667, an alloy of hard steel as the contact surface permitting the method. The blade is fastened with the dovetail fastening at the entrance edge of the blade, as shown in any single

without disturbing the others. To minimize the risk of a bursting wheel, the disk is cut thin, and the idea that the latter will fly off in small pieces if the wheel will burst from overspeeding.

Two other designs using the dovetail fastening for force-conditions (due to speed) less severe than the Laval turbine, are shown in Figs 668 and 669. They are made from drawn and polished bars of



the most important difference between the two schemes is seen in the method of spacing the blades. Distance-blocks, cut from a

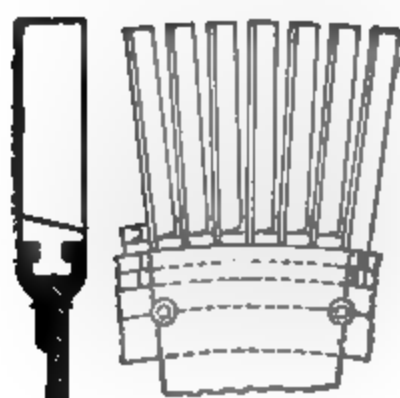


FIG. 668.—Zoelly Blading.

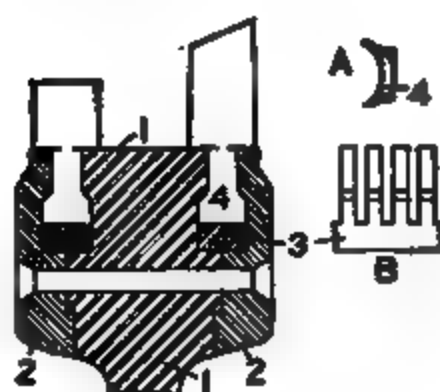


FIG. 669.—Sulzer Blading.

properly-shaped bar and machined with the blades so as to fit the dovetail slot, are used in the first case. In the second there is a spacing-ring, shown especially at B: the roots of the blades are hot-pressed and flattened, as best made evident at A, and will then slip into sawed slots in the ring 3.

The use of blades formed from rolled plate by bending and pressing is seen in Fig. 670, together with the free use of rivet-fastening, and with a shroud-ring to cover the outer ends of the blades, spacing and steadying them. Essentially the same construction is used also for the distributors of the Rateau turbine.

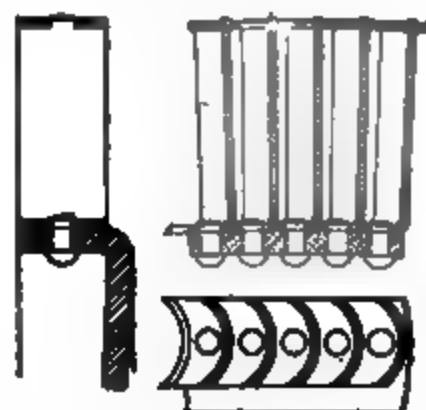


FIG. 670.—Rateau Blading.

In some designs of the Curtis turbine the blades are cut from the solid metal of the wheel, as shown at I. in Fig. 671. The cutting-tool travels on a circular path, the tool-bar either turning continually or having a back and forth rotation; in either case, this bar must be drawn backward in the direction of its axis during the idle part of the movement, so that the tool will clear, then advanced for the cutting-stroke. This method does not, however, give a very effective profile to the vanes, as can be seen by comparison with Fig. 649. It is much more usual to make the blades from metal

bar of proper shape, and two schemes for holding them are sketched at II. and III. In the first case the blades are cast into the segments of base-ring, being held in a core-sand facing which forms one side of the mold; they are made of a brass composition soft enough to be pressed into solid union with the ring. Sketch III. shows a simple dovetail fastening, with distance-blocks as in Fig. 668, and very easily applied when the holding-rings are made in a number of short segments, as is the usual practice.

The method which has been most used for holding the blades in the Parsons turbine is sketched in Fig. 672. The blades are of uniform length and simply set into the grooves, with distance-blocks of proper profile between them. These blocks have parallel sides

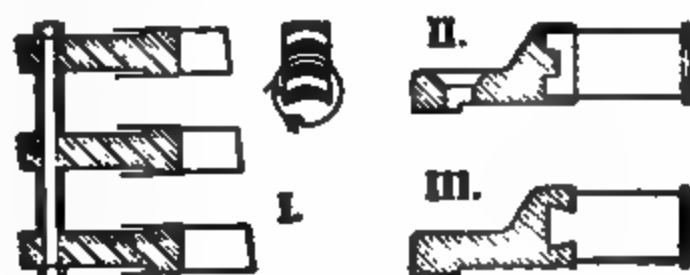


FIG. 671.—Curtis Blading.

(vertically), but are made of a soft metal; and after a whole row has been filled, the blocks are calked, by means of a tool which reaches down between the blades, and are thus pressed out, more or less perfectly, into the dovetail slot. To give a better grip to the blade, a couple of notches are made in the back of it, into which the metal of the block will be forced. The blades in the casing are held in the same way, except that plain slots are used, not dovetailed. The ends of the blades are left free, but the very long blades of the low-pressure stages must be braced against the tendency to get into vibration. One scheme is to cut a slot in the entrance sides and solder in a wire ring, as at A in Fig. 672 to obviate undesirable effects of expansion this ring will be made of several separate segments, instead of being continuous all the way around.

The scheme shown in Fig. 673, used in this country by the Allis-Chalmers Company, has a spacing and holding ring, with a block of special form into which the ends of the blades fit: these ends are

pressed into dovetail form, so that the blades are securely held. A narrow locking-ring presses the main ring into the dovetail; and on the rotor R this extra ring is calked into a little side groove. The shroud-ring, of channel-bar section, holds the blade-ends securely, and can be accurately turned off, so as to make the running-fit uniform and as close as is permissible, this minimizing leakage.

(i) **WEAR ON VANES.**—The blading is the weak point in a turbine, because it is the part most liable to rapid wear and deterioration. A steam-jet of very high velocity exerts considerable erosive action, greater if the steam is wet than if superheated, and especially aggravated when particles of solid matter are present, as when boilers prime on dirty water. Precise information along

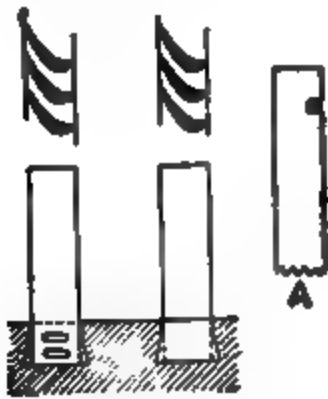


FIG. 672.—The Standard Parsons Blading.

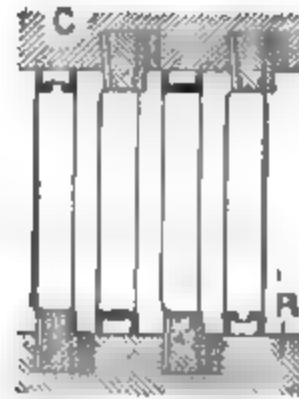


FIG. 673.—Willans-Robinson Blading for Parsons Turbines.

this line is rather scarce, and it remains to be seen how these working-parts of the turbine will compare in durability with the cylinder- and valve-surfaces which occupy a similar position in the economy of the engine. The inherent probability and the weight of experience over the short period during which turbines have been in use seem to be in favor of the engine.

The materials used for blades or vanes range from various brass or bronze compositions, through common low-carbon steel, to a nickel-steel with so high a proportion of nickel that it will be practically rustless. The Parsons type with its tremendous number of blades calls for material that can be easily worked, and with its low steam-velocity permits a comparatively soft material.

The blade-bars, of a suitable alloy (yellow metal), are generally made by the "extruding" process, semi-fluid metal being forced out through a die and cooled to a solid as it issues—after the manner long used in the manufacture of lead pipe. The turbines with fewer blades and higher velocities require harder materials but can stand a higher cost per unit part.

(j) GOVERNORS AND CONTROLLING-VALVES.—For the first example under this heading we will consider the apparatus belonging

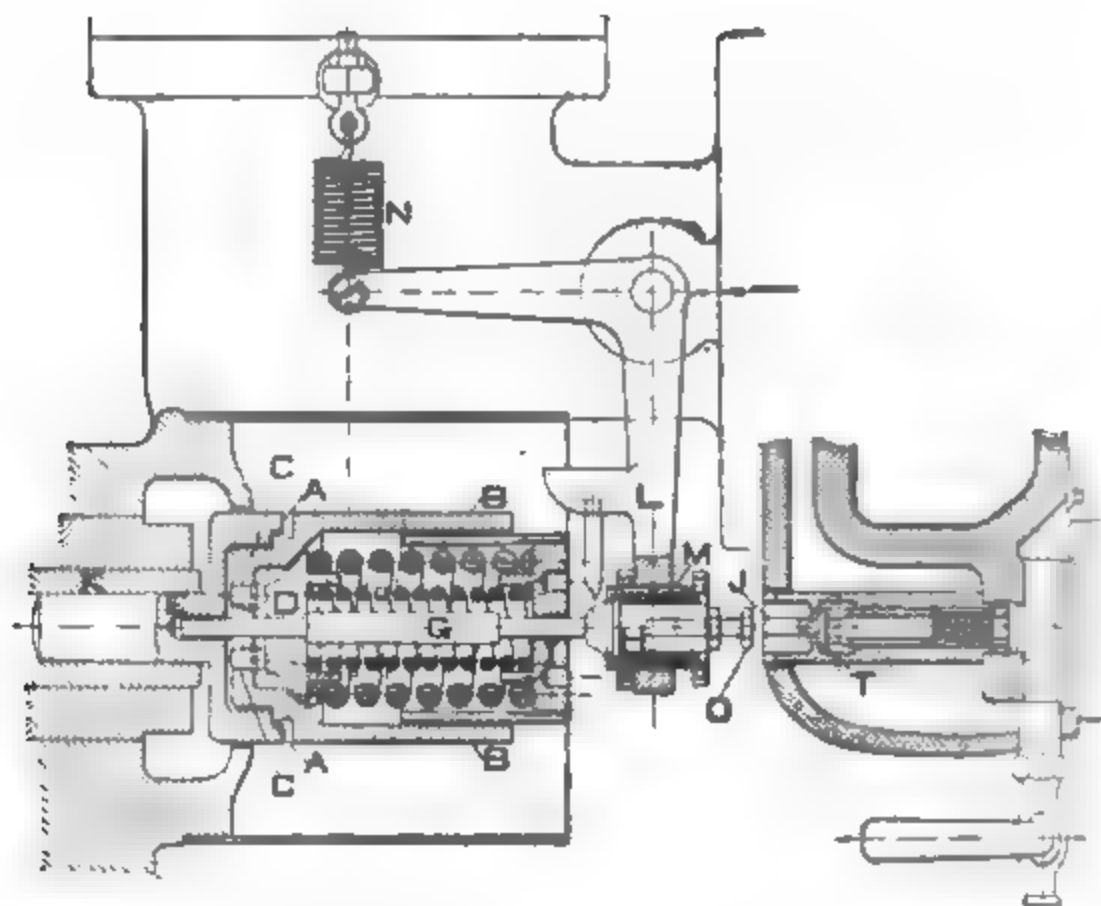


FIG. 674.—De Laval Governor and Vacuum-breaker.

to the De Laval turbine, illustrated in Figs. 674 and 675. An external view of the governor appears at H in Fig. 617, which shows how it is carried on the end of the power-shaft. The centrifugal weights, hollow semi-cylinders in form and marked B on Fig. 674, have knife-edge bearings at A, A, in the shell C. As the weights swing out, they push the block D to the right against the spring, the fixed abutment I for the latter being screwed



the outer end of C. In normal running the spindle G moves the lever L, against the light spring at N, and thus raises and lowers the double-disk throttle-valve shown in Fig. 675. Since the valve may not be tight even when down on its seat, additional security against a runaway is provided for in the vacuum-breaker. At an excessive speed the spindle G can move H in the lever L, compressing the spring M until D pushes J inward; this opens the valve at T, admitting air to the exhaust-chamber, and thus checking the wheel by friction.

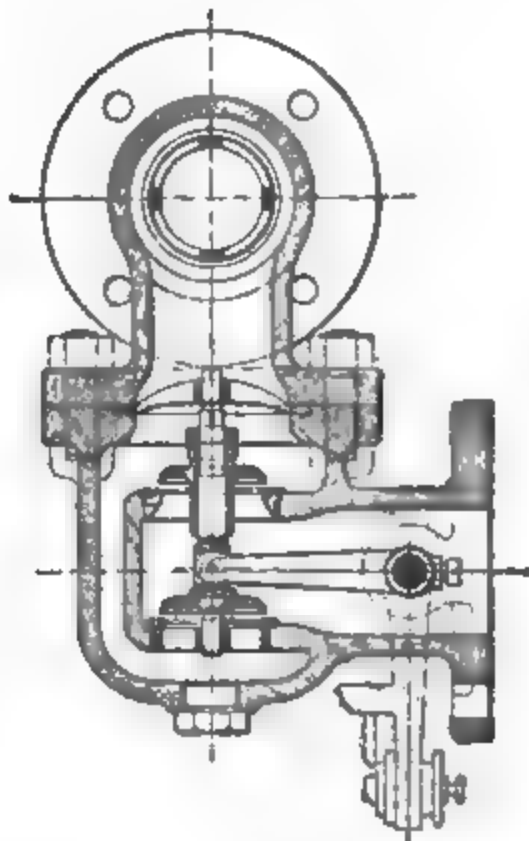


FIG. 675.—De Laval Governor-valve.

The scheme of hand-regulation, whereby the number of nozzles in service is accommodated to the load, has been described in § 71 (c), under Fig. 617; and a self-acting nozzle-valve of recent introduction is given in Fig. 679.

*The Parsons Valve and Governor.*—Fig. 676 shows the detail of the main controlling-valve of the Westinghouse-Parsons turbine, corresponding with  $V_1$  in Fig. 629. This valve is not held fast by the governor at the particular height which will make the opening just large enough, but is given a continuous oscillating movement, so as to admit the steam in puffs. The governor mechanism, outlined in Fig. 677, moves the little pilot-valve F, which opens and closes the exhaust-port E. When E is closed, steam coming through A lifts the piston C, opening the valve; when E is open the steam escapes from the cylinder more rapidly than it can get past the adjusting valve B, and the spring H pushes the valve down. Piston G has a dash-pot action, while lever K is for opening the valve by hand, so as to prevent sticking fast when the turbine is standing idle. Under light load the valve  $V_1$  is shut during the greater part of the oscillation-period; with heavy load the oscillation may entirely disappear because valve F will not open.

port E at all. The steam passing through this controlling system need not be wasted, but can be returned to the turbine at

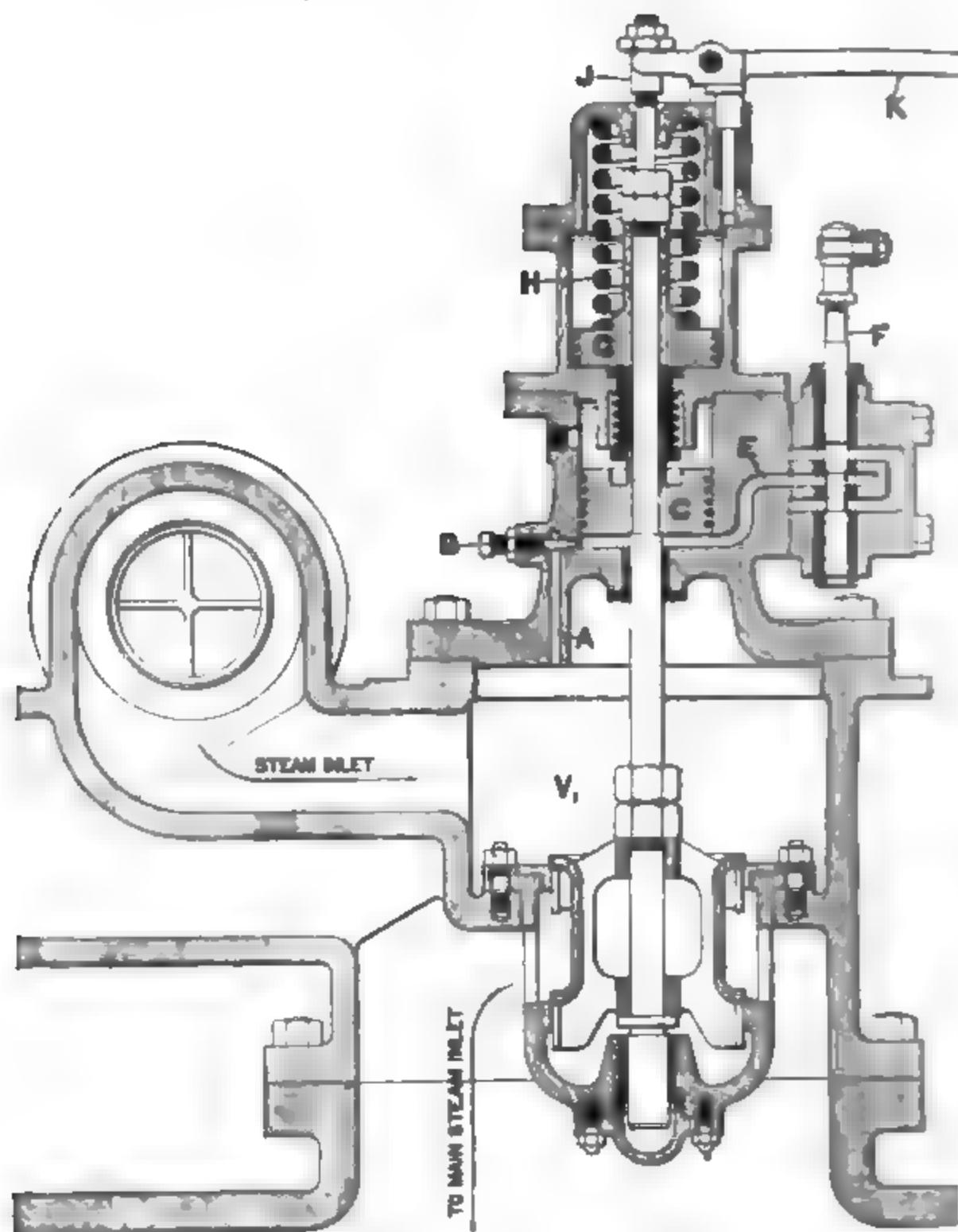


FIG. 676.—Parsons Admission-valve and Controlling-mechanism.  
Compare Fig. 629.

lower-pressure point. The by-pass valve for overload, *V*, on Fig. 629, is actuated by steam in the same manner, but

its pilot-valve is given only a simple displacement, without oscillation.

The governor-mechanism is sketched in Fig. 677. The eccentric E is on the same spindle with the wheel driven by the worm shown in Fig. 663, and the lever 1 is oscillated once to every so many revolutions, say 5 or 6, of the rotor. The governor being at one end of the machine, piece 3 takes the form of a long rod or shaft, with arms keyed to it. The function of the governor is simply to raise and lower the fulcrum A.

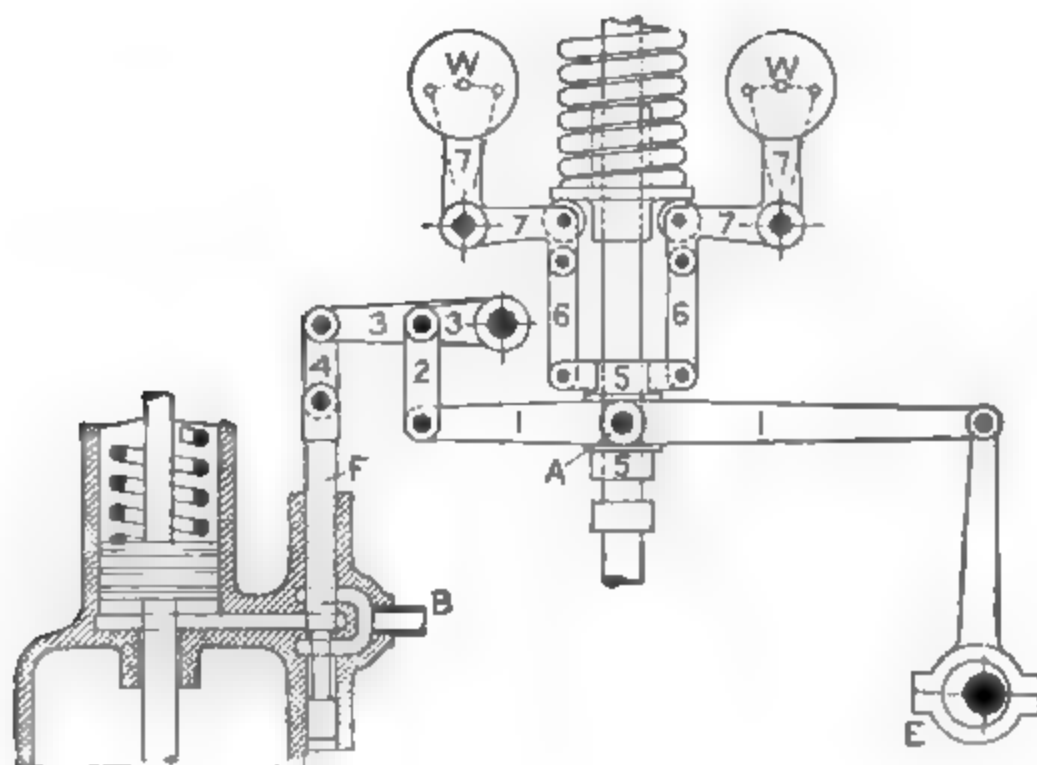


FIG. 677.—Outline of Governor and Valve-gear, Westinghouse-Parsons Turbine.

This same scheme, with variations in mechanical detail only, is used on the various makes of Parsons turbines, and has been adopted by Sulzer Bros. also, according to Stodola. Other inventors, as Rateau and Zoelly, have used plain throttling. It does not appear that puff admission has any very great advantage over steady throttling. Of course, most of the steam goes through when the pressure is high, and works therefore under the conditions favorable to good performance; but to neutralize this there are

alternate accelerations and retardations of the whole steam-current which must be harmful. No statements or diagrams have been published as to whether and in what degree these pulsations extend through the turbine to the low-pressure stages. One merit of this scheme is that the valve is never likely to stick fast so that the governor will lose control of the turbine.

*The Curtis Admission-valve.*—As stated in the general description, § 71 *g*., the Curtis turbine is controlled by the opening of

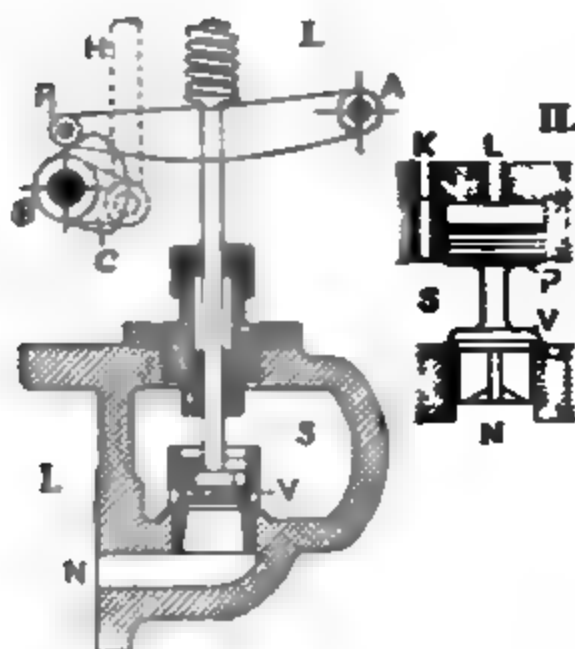


FIG. 678.—Nozzle-valves for the Curtis Turbine.

more or fewer of the nozzle-valves.

The scheme used on the machine in Fig. 621 is shown in Fig. 678 I. The governor operates a

hydraulic cylinder which is essentially equivalent to the device in Fig. 562, self-centering-valve and all, but with the piston moving

From the cross-head comes the rod H, which turns the cam-shaft B; and the cams along this shaft

one to a valve, are set in such a way that each a little later than the preceding one, or a certain number

of degrees behind it. Under a particular load there will be a

number of valves wide open, one partly open, the rest closed. The single valve that is just in the act of opening for any position of the

cam-shaft gives the close gradation of power.

In most Curtis turbines, however, steam-actuated valves have been used, according to the scheme at II. in Fig. 678. The governor moves a series of small pilot-valves, one of which determines whether steam of full pressure shall be admitted above the piston P or whether this space shall be opened to exhaust. In the latter case the valve will be lifted, because the piston is larger in diameter than the valve-disk. The pilot-valves are generally moved by a cam-shaft like that here used for the main valves; and magnet-lifts have also been employed.

*Self-acting Valves.*—In Figs. 679 and 680 are given a couple of examples of valves operated by pressure-differences which arise within the turbine. Fig. 679 is a recently adopted scheme for the De Laval turbine. With full pressure in the valve-chamber, the grooved plunger is pushed to the left against the spring; but when the governor throttles the steam, the spring-force predominates and closes the valve. The inter-stage valve in Fig. 680 has already been explained, as to purpose and action, under Fig. 621.

(k) *EFFECT OF GOVERNOR-ACTION.*—There are two typical methods of governing the tur-



FIG. 679.—Self-acting Nozzle-valve for De Laval Turbine.

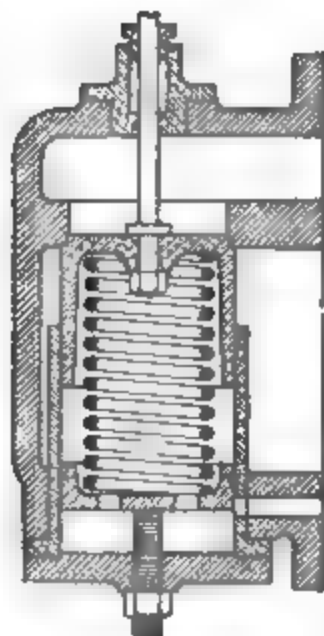


FIG. 680.—Automatic By-pass Valve for Curtis Turbine; enlarged from Fig. 621.

bine, the first by throttling (with puff-admission as a special phase), the second by varying the number of nozzles open to steam. The question presents itself, What is the effect of a change from normal load, and is the effect different under these two schemes? As characteristic of normal-load running, we assume that in all the stages the steam-velocity bears the same ratio to the vane-speed, according to § 72 (j), and that the final drop to exhaust-pressure just gives to the last stage its proper share of energy. The chief determinant of this action is the manner of variation of the channel-section, as pointed out in Art. (a), and at each point along the channel there will be a characteristic normal pressure, with which the local pressure under changed conditions may be compared.

The most obvious result of a decrease in the amount of steam admitted is to lower the local pressure all along the line, while an increase will raise it. This will give the last stages a smaller proportion of the total energy-development at light load, a larger share at heavy load. Under very light load the exhaust-pressure tends

to creep up into the turbine, the gradient through the last stages being only enough to maintain the flow; under heavy load, the pressure backs up toward the entrance.

Now plain throttling, which cuts down the steam-pressure at entrance, tends to keep the division of work among the stages more nearly equal at light loads—while diminishing, of course, the total energy available. The cut-off method admits steam at full pressure, but it has a big drop through the initial stage, and thereafter the local pressure will be about the same as with the first manner of control. The advantage of the second method depends upon the ability of the first stage to absorb effectively the relatively large energy made available for it; but since such a capability will always exist to some degree, it seems that the scheme of having all the pressure-drop within the turbine itself is inherently better than that of allowing a considerable part of the drop to take place in the governor-valve. Puff-admission acts in somewhat the same way as cut-off control, in that most of the steam enters at high pressure as just pointed out under Fig. 577. It has been suggested that control valves be applied to the successive nozzles in few-stage turbines of the Curtis type, but this idea has not yet been extended beyond the one interstage valve on Fig. 621. The intermittent action of the Parsons type requires the fly-wheel effect of the heavy rotor for the maintenance of steady speed.

## APPENDIX TO CHAPTER XII.

### THE ROTARY ENGINE.

AN immense amount of invention and ingenuity has been expended upon the problem of devising a machine in which the steam-pressure cycle could be applied to a rotating instead of a reciprocating working-element. Inherent difficulties of the first magnitude have kept these efforts from success, while the advent of the turbine has satisfied the demand which they were intended to meet. A description of the various rotary engines that have been patented would belong rather to the study of mechanism than to practical steam-engineering; and to illustrate the general principles of some we shall use but the one simple type outlined in Fig. 622.

In this machine steam enters continually at S and escapes at E, only a plain throttle-valve in the inlet-pipe being needed to control the flow. Sliding vanes or pistons, P, P, P, project from the rotor R and touch the casing C, furnishing the moving surface upon which the working steam-force acts. Some simple con-ivance, such as a circular groove in the ends of the casing and projections on the vanes, must be added in order to keep the latter out against the casing.

This particular scheme has the fault that it makes absolutely no provision for expansion of the steam below the inlet pressure, which puts it in a class with

the small direct-acting steam-pump. Better inventions have met this requirement to some degree, although far less effectively than the piston-engine. But the great and determining drawback has been the inability to hold steam, or the tendency of the machine to excessive leakage. The circular piston, moving in its cylinder with a broad contact, is the simplest possible working-element for the static-pressure cycle, and it is hard enough to keep the piston tight. The rectangular vane, with narrow contacts and variant rubbing-speed at different points of its perimeter, cannot be made and kept steam-tight; and there is a further chance of leakage between the ends of rotor and casing. As regards thermal wastes, it is evident that a considerable portion of the metal surface which the steam touches is exposed alternately to high and low temperatures. Further, since the machine develops its power by the combination of a small driving-force with a high velocity, the rubbing-friction due to an attempt to keep the piston tight will absorb an excessive amount of power. These unavoidable bad features have completely overshadowed the advantages of compactness and freedom from shaking-force.

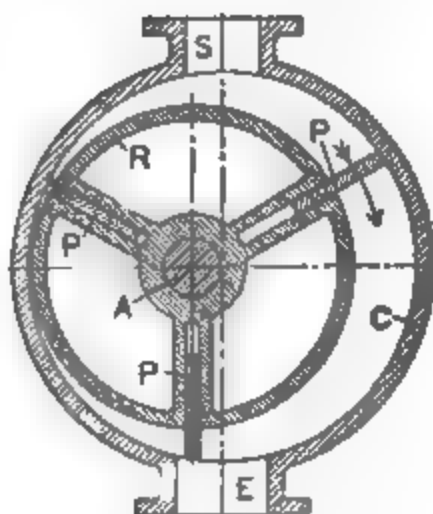


FIG. 681.—Outline of Typical Rotary Engine.

## CHAPTER XIII.

### STEAM-ENGINE PERFORMANCE.

#### 75. Steam-test Tables.

List of tables in § 75:

A. Tests of Simple Engines.....	Pages 608 to 611
B. Compound Engines, High-speed Type.....	Pages 612 to 615
C. Multiple-expansion Engines, Corliss Type. .	Pages 616 to 619
D. Pumping and Marine Engines.....	Pages 620 to 623
E. Tests of Steam-turbines. ....	Pages 624, 626, 627
F. Engine and Generator Efficiencies. ....	Page 625
G. Mechanical Efficiency of Engines. ....	Page 628

(a) THESE TABLES set forth, in condensed and concentrated form, the results of a large number of tests of engines and turbines covering the whole range of steam-engineering practice. Published reports vary widely as to fullness of detail, but here only the more important data are given, such as are available in almost every case. It will be noted that the derived results on the last page of each table are concerned wholly with the thermodynamic side of the subject. Fuller information, having to do partly with the engine alone, partly with the other members of the plant also, would lie along several lines, somewhat as follows:

- I. On the mechanical side of the steam-action,
  - (a) Pressure-losses from boiler and to condenser;
  - (b) Receiver-pressures and losses between cylinders;
  - (c) Division of work among the several stages.
- II. On the thermodynamic side,
  - (a) Indicated steam-consumption and cylinder-condensation;



- (b) Jacket pressures, and the different amounts of jacket-discharge;
- (c) With superheated steam, temperatures at various points in the progress of the steam;
- (d) Temperatures of hot-well discharge, of jacket-discharge, of condensing water, and of feed-water.

### III. As to the performance of the machine,

- (a) The mechanical efficiency, to the output point.

Division I. is best covered by the combined diagram, which also illustrates II. (a)—see § 69. The ratio of indicated steam to actual was the last item dropped when compressing the main tables into the limits of space available; some representative values are given under the figures in § 69. The rest of Group II. is desirable when a close study of the thermodynamic detail is to be made, but is not necessary for a determination of total or resultant performance. Supplementary data as to III. will be found in Tables F and G.

(b) THE TABLE-HEADINGS.—The first two pages of each table give general information about the engine and the source of data. The cylinder-dimensions are in inches, nominal sizes; metric dimensions are expressed to the nearest tenth of an inch. The “Ratio” is that of the largest to the smallest cylinder-volume (or stage-volume), passing over the intermediates in a triple or quadruple engine. Similarly, the clearances given are those in the first and last cylinders when there are more than two stages. The “Year” shows when the test was made; the authority, under “Reported by”, is generally the person who conducted the test; a dash after the name shows that other persons joined in the presentation. The abbreviations under “Reference” have the following meanings:

A. S. M. E. <i>Transactions American Society of</i>		
	<i>Mechanical Engineers,</i>	Vol. — page
ENG.	<i>Engineering</i> (London) Half-year	vol. — page
ENG'R	<i>The Engineer</i> (London) Half-year	vol. — page
E. R.	<i>Engineering Record</i> (New York)	Date — page

- STOD. Stodola's *Steam Turbines*,  
 II. E Second Edition, English.  
 III. G Third Edition, German.  
 Z. V. D. I. *Zeitschrift des Vereins deutscher Ingenieure*,  
 Year - page.

The letter *q* before a reference means that this account is quoted from some earlier publication.

(c) GENERAL CONDITIONS, POWER, AND STEAM-CONSUMPTION.\*

SECTION 1 OF ENGINE TABLE.

No. CONDITION	<i>N</i>	<i>V</i>	<i>P</i> <sub>1A</sub>	<i>P</i> <sub>2A</sub>	<i>e</i>	<i>r</i>	<i>P</i> <sub>3A</sub>	<i>H</i>	<i>S</i> <sub>3</sub>	<i>I</i>
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(For differing portion of Turbine Table, except CONDITION, see Art. (e), page 606.)

No. A number is given to each engine; the decimal figures designate different tests of the same engine.

CONDITION. By a condensed notation a number of important general conditions of the test are stated, as follows:

First letter—condition of steam:

T=saturated steam;

P=superheated steam.

Second letter—kind of exhaust:

A=atmospheric exhaust, or non-condensing;

C=condenser or vacuum exhaust.

Numeral—number of stages in the expansion.

Third and fourth letters—condition as to jackets and heaters:

J=jackets in use;

H=reheater in receiver in use;

N=no jackets or heaters.

If either J or H is given alone, absence of the other type of heating device is implied. Subscripts under indicate that only certain cylinders are jacketed—1 for first cylinder, 2 for second, etc.

\* Discussed in § 76 (a) to (e).

Last letter—manner of measuring steam-consumption:

F=feed-water weighed or measured;

M=feed-water metered;

C=steam condensed and weighed.

In Table E, for the Steam-turbine, the first two letters and the last under CONDITION have the meanings just given; the third letter shows the manner of loading:

B=brake loaded;

E=electrically loaded.

(Note.—To facilitate the reading of reports in the metric system, the reduction ratios of the important units are given under the definitions.)

Speed in revolutions or double-strokes per minute.

Piston speed in feet per minute.

(1 meter per sec. = 196.7 ft. per min.)

Initial pressure in pounds per square inch absolute.

In most cases this is the pressure at the boiler; it should be, and where known is, the pressure at (that is, just before) the throttle-valve.

(1 kg. per sq. cm. = 14.22 lbs. per sq. in.)

Absolute pressure of the exhaust, either of the atmosphere or in the condenser, in pounds per square inch. In the locomotive or when the exhaust-steam is used for heating, this pressure will be above atmosphere. See § 76 (b).

In reducing vacuum readings, 1 lb. per sq. in. = 2.035 ins. of mercury

(57.1 mm. of mercury = 1 lb. per sq. in.)

Apparent cut-off in high-pressure cylinder, or fraction of stroke completed at cut-off, not counting the clearance.

Ratio of expansion, or ratio of the final full volume of the low-pressure cylinder to the volume back of the high-pressure piston at cut-off, clearance being taken into account. If  $R$  is the cylinder-ratio and  $i_1$  and  $i_2$  the clearances as given above,

$$r = \frac{R(1 + i_2)}{e + i_1} \quad . \quad . \quad . \quad . \quad . \quad . \quad (441)$$

- $p_m$  The mean effective pressure reduced to the low-pressure piston after the usual manner—see § 23 (c), Eq. (111).
- $H$  Indicated horse-power.  
(1 metric H.P. = 0.9863 English H.P.)
- $S_H$  Steam per horse-power hour, in pounds; this is intended to be the actual steam-consumption, not corrected for moisture or superheat and including all steam used in jackets or reheaters.  
(1 kg. per met. H.P. = 2.235 lbs. per Eng. H.P.)
- $S$  Total steam used, pounds per hour—in the plots of test, Figs. 688 to 698.
- $J$  The fraction of the total steam that is used in the jackets and reheaters. A dash placed in this column indicates the absence of this quantity from the test.

## (d) THERMODYNAMIC PERFORMANCE.\*

## SECTION 2 OF ENGINE AND TURBINE TABLE.

	$m$	$t_s$	$t_i$	$Q$	$W$	$w$	$W_R$	$E_A$	$E_R$	$\eta$
$m$	Fraction of moisture in steam received by engine.									
$t_s$	Degrees (Fahrenheit) of superheat, at engine when so given; otherwise at superheater.									
$t_i$	Ideal feed temperature, or steam-temperature corresponding to the exhaust-pressure $p_{2A}$ . It is assumed that with a complete and perfect feed-heater the engine could deliver feed-water at the temperature of the exhaust-steam.									
$Q_H$	Number of heat-units (B.T.U.) added to the feed at $t_i$ by the hot water from the jackets and other heaters, or the steam contained in the weight $J$ (fraction of one pound) of feed-water over and above its heat at $t_i$ . In most cases the water from the jackets is supposed to have the full temperature $t_i$ , corresponding to the pressure $p_{1A}$ ; in the more complex engines some of the heaters have lower pressure.									

\* Discussed in § 76 (f) to (m).

**Effect of moisture or superheat on heat of steam-formation.**  
**With moisture-fraction  $m$ ,**

$r_1$  being the latent heat at  $t_1$ . With superheat  $t_s$ ,

**For value of the specific heat  $c$ , refer to § 76 (f).**

The effective work per pound of steam, measured in heat-units, or the heat transformed into useful work per pound of steam. One horse-power-hour being  $1,980,000 \div 778 = 2545$  B.T.U.,

**Correction to be made in output, with Rankine Cycle, on account of moisture or superheat in steam received by engine. See § 73 (a), Eqs. (430) to (434), and § 76 (j).**

**Effective output of Rankine Cycle, per pound of steam of actual quality, measured in heat-units.** See § 73 (a). It is the value from Table XII for  $p_{1A}$  and  $p_{2A}$ , corrected by  $w$ .

**Absolute thermodynamic efficiency for the engine alone, not including the (actual) feed-heating system. It is the ratio of effective output of the steam-cycle to the heat received, so that**

**Relative thermodynamic efficiency, with the Rankine Cycle as the standard of performance:**

$$E_R = W \div W_R. \quad . \quad . \quad . \quad . \quad . \quad (446)$$

$Q_M$  Heat supplied per horse-power-minute, which is a frequently used measure of engine performance. Since  $33000 \div 775 = 42.42$  B.T.U.,

$$Q_M = 42.4 \div E_A \quad . \quad . \quad . \quad . \quad . \quad (47)$$

(1 calorie = 3.968 B.T.U.)

1 cal. per met. H.P. = 4.024 B.T.U. per Eng. H.P.)

#### NOTE.

Where the data are printed in italics it means that the values were not determined (or given), but have been assumed as a basis for calculation.

A blank in any column shows that the quantity was not determined; except that a dash under  $J$  and a blank under  $q$  indicates the absence of jacket-action from the test.

Many of the values in the tables were computed with one more decimal place than shown, notably  $Q$  and  $E$ , but the numbers given are amply accurate for any practical significance, especially in view of the degree of precision of the data from which they are derived.

(e) EFFICIENCY DATA.—The performance of the piston-engine Tables A to D, is all expressed in terms of indicated rather than effective or brake horse-power; the latter is commercially the more important, but in the majority of the tests was not determined. When we come to the turbine, however, only the effective power, sometimes brake horse-power, more often electrical output can be directly measured. To get a basis of comparison it is necessary to reduce this to the work of the steam on the rotor by assuming efficiencies. In Table E this is done, while a lot of data for engine-generators is given in Table F, which is sandwiched between Table E as being most intimately connected with the latter. No further data as to mechanical output is gathered into Table G.

The symbols used in Tables E, F, and G, and not already defined, are as follows:

- $B$  Brake horse-power, or effective mechanical output of engine.
- $D$  Dynamometer horse-power (for the locomotive).
- $K$  Electrical output in kilowatts.
- $P$  Pump horse-power.

$E_M$	Mechanical efficiency, $B/H$ .
$E_E$	Electrical efficiency, of generator alone.
$E_C$	Combined efficiency of engine and generator, $E_M \times E_E$ .
$E_P$	Efficiency of pumping-engine, $P/H$ .
$K/H$	This ratio is the product of $E_E$ and the electrical horse-power factor 0.746, and serves for the quick comparison of Tables E and F.
$S_B$	Steam per brake horse-power-hour.
$S_K$	Steam per kilowatt-hour.

In Table E these last two are given in the same column, as either one or the other was determined; and the letters B and K are placed after the numbers to show which quantity is meant.

(f) RANGE OF TEST DATA.—Glancing over the “general information” part of the tables we see that both chronologically and geographically the field of steam-engineering has been pretty well covered. There is little data from French sources, but this lack is the less important because of the quite general use of the Corliss engine in France, so that there would only be a paralleling of English and American results. In selecting from the great mass of material available, the object has been to choose representative tests; in most cases performance at or near rated power is given, the manner of variation of efficiency with load being left for illustration by curves from several typical series of tests, in Figs. 687 to 698.

A few salient points in regard to the source and character of information as to engines of different classes are as follows:

Tests of simple engines are mostly rather ancient history, later attention being chiefly paid to the more complex and economical forms. For the small high-speed engine very little good data is available, considering the large numbers of these engines in use.

By far the best lot of work on the locomotive is that covered by the report of the tests made on the Pennsylvania Railroad Testing Plant at the St. Louis Exposition of 1904. These results are, however, a little better than those generally got in road tests.

[To page 629]

TABLE 75 A.

TESTS OF

No.	DIAMETER X STROKES			CLEARANCE	TYPE	SERVICE
SIMPLE MARINE ENGINES						
101	36	X	96	.058	Duplex, } inclined,	{ Paddlewheel
102	58	X	105	.085	Single, } lift-valve	
103	26	X	36	.054		
104	25	X	24	.041		Screw
CORLISS ENGINES						
111	8	X	24	.08	Horizontal	Experimental
112	9	X	36	.076	Horizontal	Experimental
113	23	X	60		Horizontal	Textile mills
114	20	X	48			
115	28	X	48			
116	30	X	72			
117	32	X	48	.035	Horizontal	Textile mill
118	25	X	48	.03	Duplex horizontal	Textile mill
119	24	X	48	.025	Horizontal	Textile mill
LIFT-VALVE ENGINES						
131	27.5	X	59.1	.03	Duplex horizontal	Textile mill
132	21.3	X	45.2	.025	Horizontal	Textile mill
133	22.6	X	45.3	.024	Duplex horizontal	Textile mill
134	16.3	X	10.4	.048	Horizontal	Shop power
DOUBLE-VALVE ENGINES						
141	17	X	50	.071	Horizontal (duplex)	Air-compressor
142	17	X	24	.023	Duplex hor. Buckeye	Power
HIGH-SPEED ENGINES						
151	7½	X	10	.118	Hor. semi-portable	Farm power
152	8½	X	15		Hor. semi-portable	Farm power
153	10	X	20		Hor. Porter-Allen	Experimental
154	10	X	11		Hor. Ball & Wood	Experimental
155	12	X	14	.088	Hor. McEwen	Generator
156	14	X	20		Hor. slide-valve	Power
HIGH-SUPERHEAT ENGINES						
171	9.8	X	6.3		Vert. piston-valve	Power
172	7.1	X	11.8		Horizontal	Power
SIMPLE LOCOMOTIVE						
181	22	X	28	.113	Consolidation	Penna. R.R.



SIMPLE ENGINES.

TABLE 75 A.

LOCATION	YEAR	REPORTED BY	REFERENCE	No.
S.S. Michigan	1861	B. F. Isherwood, "Researches in Steam Engineering"	Vol. I. 96	101
S.S. Eutaw	1863		Vol. II. 144	102
S.S. Dexter	1874	C. E. Emery	Eng. 1875, I. 129	103
S.S. Bache	1874		Eng. 1875, I. 2	104
Mass. Inst. Tech.	1885	C. H. Peabody	A. S. M. E., 7-328	111
Sibley College	1894	R. C. Carpenter	A. S. M. E., 16-913	112
New England cot- ton mills	1880	G. H. Barrus	A. S. M. E., 11-175	113
	and ear- lier			114
				115
				116
	1879	N. McDougall	Eng. 1881, II. 35	117
Paisley	1889	M. Longridge	Eng. 1892, II. 557	118
Mulhouse	1878	Soc. Ind. de Mulhouse	Eng. 1881, I. 347	119
Augsburg	1872	C. Lindé	Eng. 1873, I. 74	131
Augsburg	1872	C. Lindé	q Z. V. D. I. 1905, 1310	132
Augsburg	1876	Prof. Thoma	q Z. V. D. I. 1905, 1310	133
Augsburg	1904	J. Krumper	Z. V. D. I. 1905, 1310	134
	1889	J. E. Denton—	A. S. M. E., 10-722	141
	1884	J. C. Hoadly—	q E. R., Apr. 7, '94-303	142
Newcastle	1887	Roy. Agric. Soc.	Eng. 1887, II. 518	151
Mass. Inst. Tech.	1885	C. H. Peabody	A. S. M. E., 7-328	153
Levens Inst. Tech.	1894	D. S. Jacobus	A. S. M. E., 15-915	154
	1893	R. C. Carpenter	A. S. M. E., 14-426	155
Onesville, Mich.	1890	R. C. Carpenter	A. S. M. E., 11-723	156
Weißenbrücken	1896	Prof. Gutermuth	Z. V. D. I. 1896, 1391	171
Wien	1894	Prof. Gutermuth	Z. V. D. I. 1896, 1423	172
St. Louis	1904	P. R.R. Report of Tests	p. 564, 580	181

TABLE 75 A.

TESTS OF

No.	CONDITION	N	V	P <sub>1A</sub>	P <sub>2A</sub>	e	r	P <sub>m</sub>	H	S <sub>m</sub>	J
101	TC1NF	13.7	219	35.8	2.12	.30	3.0	19.9	134	35.2	—
102.1	TC1NF	9.2	161	38.7	0.84	.58	1.6	32.5	415	31.39	—
2	P	9.1	159	41.9	1.45	.50	1.9	31.5	397	26.15	—
103.1	TC1NF	61.1	367	81.9	2.3	.25	3.5	37.5	219	23.91	—
2		50.9	305	55.5	2.0	.26	3.3	25.6	124	28.80	—
104.1	TC1NF	47.1	188	92.7	2.74	.16	5.3	32.3	89	26.25	—
2	J	53.8	215	94.1	2.12	.16	5.1	36.9	116	23.15	(6)
111	TA1NC	61.3	245	86.6	14.7	.37	2.4	45.8	18	36.3	—
112.1	TC1NF	85.4	512	115.7	2.7	.20	4.3	54.7	52	23.5	—
2	J	85.4	512	113.1	2.2	.17	4.8	57.7	50	22.9	(65)
113	TA1NF	74.7	747	87.0	17.5	.37	2.6	33.1	309	27.8	—
114	TC1NF	61.7	502	99.4	2.6	.18	5.0	41.8	195	19.24	—
115	PA1NF	100.0	800	83.5	17.5	.33	2.9	28.6	423	25.53	—
116	PC1NF	50.1	601	84.2	2.7	.18	5.0	29.2	373	18.48	—
117	TC1JF	82.8	662	85	1.4	.21	4.3	33.8	540	20.7	(6)
118.1	TC1NF	65.0	520	76	2.0	.22	4.2	33.2	508	19.77	—
2	J	65.1	521	76	2.0	.22	4.2	31.9	488	19.27	(6)
119.1	TC1NF	49.2	394	80.6	2.1			24.2	127	24.73	—
2	J	50.2	402	91.8	2.1			29.9	161	19.31	(6)
131	TC1JF	49.3	371	95.9	1.18	.10	8.4	26.5	105	19.35	—
132	TC1JF	39.6	389	89.8	2.6	.10	7.7	28.7	395	19.29	(6)
133	TC1JF	47.7	360	93.5	1.37	.12	7.4	30.2	262	18.11	—
134.1	TC1JF	81.5	535	159.9	1.18	.05	10.3	37.7	127	17.21	—
2	P	81.3	533	159.5	0.71	.06	9.5	37.0	121	16.68	—
141.1	TA1NC	59.9	300	105.7	14.4	.23	3.6	45.8	91	27.41	—
2		9.3	47	107.2	14.4	.15	4.8	40.1	12	38.22	—
3		65.7	324	43.3	14.4	.22	3.7	9.8	21	52.11	—
142	PA1NF	152.9	612	90.6	14.7	.34	2.8	36.8	311	25.64	—
151	TA1JF	172.3	287	135	14.7	.08	6.0	37.0	14	25.4	(6)
152	TA1JF	136.3	340	140	14.7	.13	4.5	40.7	23	20.4	(6)
153	TA1NC	198.7	660	88.5	14.7	.27	3.1	21.8	34	43.4	—
154	TA1NC	253.9	465	85.7	14.7			25.6	28	37.7	—
155	TA1NC	245	571	92.1	14.7	.32	2.7	38.8	76	31.1	—
156	TA1NF	171.5	572	85.0	14.7	.30	3.0	17.2	46	30.3	—
171.1	PA1NF	250	262	119.9	14.7			27.7	17	19.4	—
2	T	250	262	117.1	14.7			24.5	15	39.6	—
172	PA1NF	200	396	142.8	14.7			48.7	23	14.81	—
181.1	TA1NF	40.4	187	217.9	15.0	.31	2.7	106.4	455	27.49	—
2		79.7	372	215.2	15.3	.29	2.8	92.4	779	24.13	—
3		120.4	561	210.4	16.3	.33	2.5	80.2	1006	23.79	—
4		159.1	742	195.8	17.4	.30	2.7	58.9	990	24.25	—
5		160.8	750	186.6	16.7	.45	2.0	47.3	803	29.33	—

For meaning of symbols, see pages 602 to 606.

MPLE ENGINES.

TABLE 75 A.

<i>t</i> <sub>s</sub>	<i>t</i> <sub>2</sub>	<i>q</i> <sub>H</sub>	<i>h</i>	<i>Q</i>	<i>W</i>	<i>w</i>	<i>W</i> <sub>R</sub>	<i>E</i> <sub>A</sub>	<i>E</i> <sub>R</sub>	<i>Q</i> <sub>M</sub>	No.
15	128		-13.9	1051	72.4	-2.5	180.8	.069	.401	617	101
15	93		-13.9	1045	81.1	-3.2	194.5	.078	.417	548	102.1
15	115		+51.5	1133	97.5	+20.8	235.6	.086	.413	493	2
15	132		-13.4	1065	106.4	-3.2	235.8	.100	.452	425	103.1
15	126		-13.7	1061	88.5	-2.9	213.0	.083	.416	510	2
15	138		-13.3	1061	97.0	-3.1	229.8	.091	.422	465	104.1
15	123	-5.5	-13.3	1055	110.0	-3.3	242.3	.103	.453	411	2
17	212		-15.1	983	70.2	-2.1	124.4	.071	.565	595	111
07	138		- 6.1	1073	108.3	-1.5	247.1	.101	.457	420	112.1
08	130	-11.6	- 7.0	1068	111.1	-1.9	249.8	.104	.445	407	2
1	221		- 8.9	980	91.6	-1.1	116.2	.094	.789	454	113
1	136		- 8.8	1069	132.2	-2.1	238.3	.124	.554	342	114
0°	221		+17.7	1006	99.7	+3.1	117.4	.100	.849	428	115
0°	138		+11.8	1084	137.8	+3.9	231.2	.127	.623	334	116
2	114	-13.3	-17.8	1066	123.0	-4.7	257.7	.115	.477	368	117
15	126		-13.5	1068	128.7	-3.2	233.3	.121	.552	352	118.1
15	126	- 6.1	-13.5	1062	132.1	-3.2	233.3	.124	.567	341	2
2	128		0	1081	102.9	0	237.9	.095	.433	447	119.1
2	128	- 9.0	0	1075	131.8	0	246.3	.123	.535	346	2
1	108	-12.5	- 8.9	1084	131.5	-2.4	276.5	.121	.476	350	131
1	103	-14.1	- 8.9	1086	131.9	-2.5	231.2	.121	.571	349	132
1	113	-12.5	- 8.9	1078	140.5	-2.4	267.4	.130	.526	325	133
1	108	-18	- 8.6	1090	147.8	-2.7	309.6	.136	.477	313	134.1
1°	91	-18	+140.	1256	152.6	+57.0	393.1	.122	.388	348	2
08	211		- 7.0	996	92.9	-1.1	143.2	.093	.647	456	141.1
08	211		- 7.0	996	66.3	-1.1	142.2	.067	.466	637	2
08	211		- 7.4	978	48.8	-0.6	71.2	.050	.685	848	3
1°	212		+24.3	1023	99.2	+3.7	135.5	.097	.732	438	142
	212	-12.7	- 8.7	986	100.2	-1.5	159.3	.102	.629	418	151
	212	-16.1	- 8.6	984	124.8	-1.5	162.0	.127	.771	334	152
	212		- 8.9	989	58.7	-1.2	128.8	.059	.457	715	153
8	212		- 7.1	991	67.6	-1.0	126.7	.068	.534	622	154
5	212		- 4.4	995	81.8	-0.6	132.3	.082	.619	516	155
	212		- 8.9	989	84.0	-1.2	125.8	.085	.668	499	156
°	212		+132.5	1138	131.1	+35.7	187.8	.115	.699	368	171.1
	212		0	1010	64.3	0	150.4	.064	.427	663	2
°	212		+188.5	1198	171.8	+57.5	222.4	.143	.772	296	172
0	213		- 8.1	1011	92.6	-1.7	192.7	.092	.482	463	181.1
1	214		- 9.6	1008	105.5	-2.0	190.3	.105	.555	405	2
2	217		- 9.7	1004	107.0	-1.9	184.8	.107	.580	398	3
1	221		- 9.0	999	105.0	-1.7	175.6	.105	.598	403	4
9	219		- 7.9	1001	86.7	-1.5	174.9	.087	.497	490	5

Italic figures mean assumed values.

TABLE 75 B.

COMPOUND ENGINES

No.	DIAMETER		X	STROKE	RATIO	CLEARANCE	TYPE
COMPOUND LOCOMOTIVES							
201	23,	35	X	32	2.33	.167-.057	2-cyl. "Consolidation"
202	14.2,	23.7	X	25.2	2.81	.133-.098	4-cyl. "Atlantic"
203	15,	25	X	26	2.81	.186-.066	4-cyl. "Atlantic"
204	14.2,	22.1	X	23.2	2.46	.117-.104	4-cyl. "Atlantic"
HIGH-SPEED ENGINES							
211	6,	9	X	12	2.28	.241-.155	Semi-portable, horiz.
212	4½,	9½	X	10	4.04	.18-.18	Semi-portable, horiz.
213	5½,	9	X	15	2.48	.160-.108	Semi-portable, horiz.
214	12,	20	X	13	2.81	.11-.08	Vert. 2-crank
215	17,	27	X	24	2.55		Vert. 2-crank
216	9,	16	X	14	3.08	.12-.09	Horiz. tandem
217	12.8,	19.7	X	14.1	2.38	.08-.05	Vert. 2-crank
SCHMIDT "MOTORS" (See Fig. 612)							
221	12.2,	27.3	X	19.8	3.92		Vert. tandem
222	14.9,	31.5	X	31.5	3.46		Vert. tandem
ENGLISH HIGH-SPEED ENGINES							
224	Willans Compound, No. 3748				} Dimensions not given	} Vert. tandem, duplex or triplex	
225	Willans Compound, No. 3571						
226	Willans Compound, 800 H.P.						
227	Willans Triple, No. 3416						
228	Belliss Triple						
LOW-SPEED SLIDE-VALVE ENGINES							
232	15,	35.8	X	50	4.75	.065-.094	Horiz. direct-exp.
233	20,	34	X	60	2.92	.083-.068	Horiz. 2-crank
ENGINES WITH POSITIVELY DRIVEN CORLISS VALVES							
234	14,	28	X	18	3.77		Horiz. tandem
235	19.7,	29.5	X	19.7	2.27	.051-.048	Vert. 2-crank
236	15,	40½	X	27	7.33	.040-.047	Horiz. tandem
ENGINES WITH DOUBLE SLIDE-VALVES							
241	14,	28	X	24	4.04	.036-.064	Hor. 2-crank Buck
242	24,	48	X	48	4.13	.025-.020	} Vertical, 2-crank McIntosh & Seymour
243	28,	58	X	48	4.30	.055-.036	
244	31,	64	X	48	4.32	.061-.081	

HIGH-SPEED TYPES.

TABLE 75 B.

SERVICE	LOCATION	YEAR	REPORTED BY	REFERENCE	No.
Electric Cent., Schenectady P. R.R., French built Co., T. & S. F., Baldwin Manchester Mach. Works			Tested at Penna. R.R. Testing Plant, St. Louis Exposition, 1904: from Report of Tests, P. R.R.	{ Pp. 285, 301 Pp. 413, 429 Pp. 484, 500 Pp. 564, 580	201 202 203 204
Tests made at Royal Agric. Soc. Show, Newcastle		1887	Committee on Tests	ENG. 1887, II. 518	{ 211 212 213
Generator Boston		1898	G. H. Barrus	E. R., Aug. 6, '98-206	214
Generator St. Louis		1904	J. R. Bibbins	E. R., Jan. 28, '05-92	215
Generator Ridgeway, Pa.		1892	R. C. Carpenter	A. S. M. E., 14 - 426	216
Shop Power Augsburg		1898	J. Krumper	Z. V. D. I. 1905, 1318	217
	Cassel	1894	M. Schroeter	Z. V. D. I. 1895, 5	221
Factory Schattau		1899	R. Doerfel	Z. V. D. I. 1899, 1562	222
Generator		1904		ENG'R, 1905, II. 79	{ 224 225 226
Generator					
Mill					
Generator Blackburn			A. S. Giles 1904 Makers	ENG'R, 1904, II. 234 ENG'R, 1905, II. 78	227 228
Mill Power Mulhouse		1876	Soc. Ind. de Mulh.	ENG. 1878, I. 4, 21	232
Textile mill Blackburn		1881	M. Longridge	ENG. 1882, I. 174	233
Generator Philadelphia		1901	Davis & Griggs	E. R., Mar. 9, '01-220	234
Shop Power Augsburg		1893	J. Krumper	Z. V. D. I. 1905, 1318	235
	Harrisburg, Pa.	1903	B. T. Allen	A. S. M. E., 25 - 212	236
Generators Chicago		1893	G. H. Barrus	E. R. Feb. 17, '94-192	241
Textile Mill Lawrence, Mass.		1897	F. W. Dean	A. S. M. E., 19 - 310	242
Generators Boston		1901	L. S. Marks	A. S. M. E., 25 - 443	243
Generators Boston		1903			244

TABLE 75 B.

COMPOUND ENGINE.

No.	CONDITION	N	V	P <sub>1A</sub>	P <sub>2A</sub>	s	r	P <sub>m</sub>	H	S <sub>E</sub>	
201.1	TA2NF	40.0	213	223.9	15.0	.46	4.0	77.3	477	20.43	—
2		80.0	427	223.2	17.2	.55	3.4	79.8	987	20.10	—
3		160.0	854	223.8	18.8	.50	3.7	38.1	941	21.27	—
202.1	TA2NF	160.0	673	229.3	15.9	.38	6.0	45.5	809	19.86	—
2		240.0	1010	230.2	16.0	.29	7.4	23.5	625	22.59	—
203.1	TA2NF	160.0	693	236.5	16.4	.51	4.3	63.5	1296	19.67	—
2		240.0	1040	232.4	19.0	.50	4.4	50.0	1529	20.81	—
204.1	TA2NF	80.0	315	216.4	15.1	.45	4.8	66.7	481	17.82	—
2		160.0	629	213.5	15.8	.44	4.9	51.7	742	17.35	—
3		239.8	942	207.8	16.6	.42	5.1	34.5	744	17.91	—
211	TA2NF	172.9	317	140	14.7	.43	3.9	35.4	21	28.00	—
212	TA2JF	159.9	266	265	14.7	.07	18.5	33.6	19	19.20	—
213	TA2JF	149.0	372	170	14.7	.16	8.6	34.3	24	20.60	—
214	TA2NF	271.2	587	181.4	15.3	.37	5.5	33.7	188	21.14	—
215	TA2NF	200	800	165	14.7			40.2	554	20.01	—
216.1	TA2NF	265	619	126.3	14.0			30.6	114	23.81	—
2	J	264.3	618	129.5	14.0			29.0	108	23.14	—
3	TC2NF	265	619	124.0	2.6			29.3	109	19.95	—
4	J	264	617	128.0	2.8			28.8	107	19.60	—
217.1	TA2JF	181.1	427	133.2	14.1	.31	6.4	32.2	126	21.85	—
2	C	180.6	426	133.6	1.05	.13	11.9	30.2	118	16.10	—
221	PC2NF	116.8	395	183.0	1.1	.34	11.0	27.3	73	10.40	—
222	PC2NF	96.2	504	151.6	1.3	.45	7.5	36.5	168	9.95	—
224	PA2NF	302		169	14.7			46.2	600	15.87	—
225.1	TC2NF			167	2.4			49.9	360	16.9	—
2	P			173	2.0			50.3	360	13.4	—
226	PC2NF	273		168	2.4				800	11.23	—
227	PC3NF			194.3	3.9			34.4	1288	11.05	—
228	PC3NF	472		175	1.6				316	10.8	—
232	TC2JF	39.7	331	72.5	1.0	.83	5.8	21.2	178	19.87	—
233.1	TC2NF	47.6	476	103.8	1.5	.29	8.4	24.1	314	16.92	—
2	JH	47.6	476	104.5	1.5	.26	9.1	24.7	326	17.68	—
234	TA2NF	200	600	152.3	14.7			28.2	314	23.5	—
235	TC2JHF	149.5	489	114.1	1.26	.19	10.0	25.7	270	15.3	—
236.1	TC2JC	155.2	698	104.4	2.7			3.2	87	14.42	—
2		150.1	675	166.7	2.0			18.4	502	12.60	—
241.1	TA2NC	165.7	663	132.5	14.7	.49	8.2	21.6	267	23.24	—
2	C	163.3	653	133.5	2.0	.33	11.9	23.5	286	16.07	—
242	PC2J,HF	99.6	797	137.6	1.3			24.9	1076	12.87	—
243.1	PC2JHM	101.9	815	164.2	1.1	.23	17.6	16.2	1047	13.39	—
2	N	101.9	815	173.6	1.0	.20	19.2	17.0	1105	14.41	—
244	PC2JHF	116.9	935	153.2	2.2	.27	16.6	24.0	2184	12.50	—

For meaning of symbols see pages 602 to 606.

HIGH-SPEED TYPES.

TABLE 75 B.

<i>n</i> , <i>lb</i>	<i>t</i> <sub>2</sub>	<i>q</i> <sub>H</sub>	<i>h</i>	<i>Q</i>	<i>W</i>	<i>w</i>	<i>W</i> <sub>R</sub>	<i>E</i> <sub>A</sub>	<i>E</i> <sub>R</sub>	<i>Q</i> <sub>M</sub>	No.
166	213		-13.9	1005	124.5	-2.9	193.5	.124	.644	342	201.1
161	222		-13.5	997	126.6	-2.7	185.1	.127	.683	334	2
164	223		-13.7	996	109.4	-2.7	179.6	.110	.610	386	3
185	216		-15.4	1002	128.1	-3.2	191.3	.128	.670	332	202.1
173	216		-14.4	1002	112.7	-3.0	191.4	.112	.589	382	2
182	217		-15.1	1001	129.4	-3.2	191.8	.129	.675	328	203.1
253	225		-21.1	987	122.3	-4.2	180.3	.124	.678	343	2
7°	213		+57.8	1076	142.8	+14.3	202.9	.133	.703	320	204.1
0°	216		+65.0	1080	146.6	+15.7	205.4	.136	.713	313	2
2°	218		+59.4	1072	142.1	+13.6	198.2	.133	.717	320	3
1	212		- 8.7	1000	91.0	-1.5	162.0	.091	.561	466	211
1	212	-23.5	- 8.2	993	132.6	-1.9	208.2	.134	.636	317	212
1	212	-17.6	- 8.5	988	123.2	-1.6	175.8	.125	.701	340	213
15	215		-12.7	1000	120.4	-2.4	177.4	.120	.680	352	214
13	212		-11.1	1002	127.2	-2.1	175.4	.127	.725	334	215
1	210		- 8.7	1000	106.8	-1.5	157.5	.107	.678	398	216.1
1	210	-4.9	- 8.7	1001	110.0	-1.5	159.4	.110	.690	386	2
1	135		- 8.7	1075	127.6	-2.2	254.1	.119	.502	358	3
1	139	-11.4	- 8.7	1072	129.9	-2.2	251.2	.121	.518	351	4
1	210	-12	- 8.7	989	116.5	-1.5	159.9	.118	.729	360	217.1
1	104	-18	- 8.7	1090	158.0	-2.6	303.6	.145	.520	293	2
3°	108		+153	1273	244.6	+62.2	386.7	.192	.633	221	221
5°	111		+161	1273	255.8	+64.7	368.8	.200	.693	212	222
7°	212		+67.4	1081	160.5	+16.0	193.0	.148	.831	286	224
	133		0	1093	150.6	0	279.4	.138	.539	308	225.1
0°	126		+93.0	1193	190.0	+32.8	323.6	.159	.588	267	2
7°	133		+61.3	1154	226.6	+19.7	299.5	.196	.756	216	226
2°	152		+72.3	1149	230.3	+22.4	286.5	.201	.804	212	227
5°	118		+187.3	1296	235.6	+77.9	381.0	.182	.618	233	228
1	102	-13.7	-9.0	1050	128.1	-2.4	265.6	.122	.483	348	232
1	116		-8.8	1090	150.3	-2.4	269.7	.138	.558	307	233.1
1	116	-15.0	-8.8	1075	149.1	-2.4	270.2	.139	.552	306	2
17	212		-14.6	996	108.3	-2.6	166.9	.109	.650	390	234
1	110	-18	-8.8	1080	164.2	-2.4	283.8	.152	.579	278	235
0	138	-17.1	0	1060	176.5	0	241.7	.167	.731	255	236.1
0	126	-10.6	0	1088	201.2	0	288.6	.185	.696	230	2
05	212		-4.3	1003	109.5	-0.7	158.7	.109	.690	389	241.1
05	126		-4.3	1090	158.5	-1.2	272.7	.145	.583	292	2
0°	111	-24.0	+12.6	1122	197.8	+3.7	311.4	.176	.635	241	242
2°	105	-19.8	+6.0	1106	187.3	+2.1	319.5	.169	.587	251	243.1
6°	102		+9.5	1134	176.7	+2.8	328.2	.156	.539	272	2
2°	130	-11.5	+40.5	1123	203.6	+12.1	290.3	.181	.701	234	244

*Italic figures mean assumed values.*

TABLE 75 C.

MULTIPLE-EXPANSION ENGINES.

No.	DIAMETERS			×	STROKE	RATIO	CLEARANCES	TYPE
CORLISS ENGINES								
251	9,	16	×	36	3.24	.076-.089		Horiz. 2-crank
252	12,	22	×	42	3.38			Hor. 2-ck. Cross
253	26,	48	×	60	3.41	.025-.025		Horiz. 2-crank
254	30,	60	×	72	4.08	.026-.036		Horiz. 2-crank
255	17,	34	×	60	4.18	.035-.027		Vert. beam
256	25,	50	×	36	4.08			Vert. 2-crank
257	20,	40	×	42	4.03	.047-.070		Horiz. 2-crank
258	25,	52	×	60	4.50	.046-.054		Vert. 2-crank
259	42,	86	×	60	4.30	.098-.048		Horiz.-vert. d.f.
HIGH-RATIO COMPOUND ENGINES								
271	44,	75.6-75.6	×	60	6.15	.104-.042		Vert. 3-crank
272	18,	44.3	×	72	6.40	.023-.018		Hor. 2-c. Wheel
273	18,	48½	×	48	7.30	.020-.029		Hor. 2-c. Wheel
274	16,	40	×	48	6.35	.025-.020		Horiz. 2-crank
LIFT-VALVE ENGINES								
281	14.6,	24.1	×	37.5	2.75	.043-.031		Horiz. 2-crank
282	26.6,	41.5	×	53.2	2.45	.040-.035		Horiz. 2-crank
283	17.2,	26	×	43.2	2.39	.044-.035		Horiz. tandem
284	28.6,	43.4	×	51.2	2.31	.050-.045		Horiz. tandem
285	26.6,	43.3	×	45.3				Horiz. tandem
HIGH-SUPERHEAT ENGINES								
291	16,	28	×	42	3.13	.041-.058		Hor. lift & Corliss
292	21,	36	×	36	2.98	.087-.094		Vert. 2-ck. piston
TRIPLE-EXPANSION POWER ENGINES								
301	9,	16,	24	×	36	7.34	.076-.094	Hor. 3-ck. } Corliss
302	11.5,	19,	30	×	36	6.90	.056-.043	Vert. 2-ck. }
303	19,	29,	46	×	48	6.00		Vert. 3-ck. }
304	17.5,	28.5,	48	×	36	7.60		Vert. 3-ck. }
311	11.1,	17.7,	27.6	×	39.4	6.63	.103-.038	Hor. 2-ck. }
312	21.7,	33.5,	51.2	×	61.1	5.85	.051-.033	Horizontal }
313	17.8,	27.6,	41.4	×	53.2	5.71	.045-.035	Horizontal }
314	15.4,	25.2,	37.4	×	25.6	6.05	.100-.053	Vert. 3-ck. }
315	23.6,	37.4,	53.1	×	35.4	5.10		Vert. 3-ck. }
316	34.0,	49.1,	61.0-61.0	×	51.2	6.40		Vert. 2-ck. }



## CORLISS TYPE.

## TABLE 75 C.

SERVICE	LOCATION	YEAR	REPORTED BY	REFERENCE	No.
Experi'tal Generator	Sibley College (Load fluctuates)	1894	R. C. Carpenter	A. S. M. E., 16-913	251
Text. mill	Fall River, Mass.	1888	G. H. Barrus	A. S. M. E., 19-320	252
Text. mill	N. Bedford, "	1893	J. E. Denton—	A. S. M. E., 9-545	253
Flour mill		1895	M. Longridge	A. S. M. E., 15-882	254
Text. mill	Lawrence, Mass	1898	Dean & Main	Eng. 1896, I. 132	255
Generator	Brooklyn, N. Y.	1902	D. S. Jacobus	Eng. 1898, II. 533	256
Text. mill	Bessbrook, Ire.	1904	E. G. Hiller	A. S. M. E., 24-1274	257
Generator	New York	1906	H. G. Stott	q Eng. '04, II. 165	258
					259
Generator	New York	1904	J. D. Andrews—	q E. R., May 28, '04-690	271
Text. mill	Grosvenordale, Conn.	1894	G. H. Barrus	E. R., Nov. 3, '94-379	272
Text. mill		1897	G. H. Barrus	E. R., N. 20, '97-540	273
Text. mill	Providence, R. I.	1902	G. H. Barrus—	E. R., Nov. 8, '02-436	274
Text. mill	Augsburg	1880	M. Schroeter	q Z. V. D. I. '05, 1314	281
Text. mill	Augsburg	1893	Bav. Br. Insp. Co.	q Z. V. D. I. '05, 1316	282
Generator	Wiesbaden	1899	J. Krumper	Z. V. D. I. '05, 1312	283
Generator	Wiesbaden	1900	Bav. Br. Insp. Co.	q Z. V. D. I. '05, 1312	284
Generator	Antwerp	1905		Eng. 1905, II. 179	285
Mill power	Philadelphia	1903	D. S. Jacobus	A. S. M. E., 25-264	291
Mill power	Belfast	1903	M. Longridge—	Eng'r, 1905, I. 546	292
Experi'tal Generator	Sibley College	1894	R. C. Carpenter	A. S. M. E., 16-913	301
Text. mill	Kimberly, S. A.	1893	A. M. Robeson	Eng. 1896, I. 129	302
Generator	Belfast	1894		Eng. 1894, II. 230	303
Shop power	Newcastle-on-T.	1901	W. B. Woodhouse	q Eng. 1902, II. 142	304
Text. mill	Augsburg	1889	M. Schroeter	Z. V. D. I. 1890, 7	311
Text. mill	Augsburg	1893	Bav. Br. Insp. Co.	q Z. V. D. I. '05, 1346	312
Text. mill	Immenstadt	1897	Bav. Br. Insp. Co.	q Z. V. D. I. '05, 1350	313
Shop power	Augsburg	1902	J. Krumper	Z. V. D. I. '05, 1352	314
Factory	Prague	1898	R. Doerfel	Z. V. D. I. '99, 1560	315
Generator	Berlin	1900		Z. V. D. I. 1900, 606	316

TABLE 75 C.

MULTIPLE-EXPANSION ENGINES.

No.	Condition	N	V	P <sub>1A</sub>	P <sub>2A</sub>	e	r	P <sub>m</sub>	H	S <sub>m</sub>	J					
251	1 TC2NC	85.1	511	112.5	3.8	.21	12.2	21.5	67	16	32					
	2 JH	85.5	513	128.9	3.2	.15	15.5	21.8	68	14	43					
252	1 TC2HF	113.	791	123.3	2.4			21.2	189	20	45					
	2 N	115.	805	128.8	2.4			19.9	181	19	34					
253	TC2NC	52	3	523	109	0	1	3	21.4	607	16	38				
254	PC2NM	65	2	783	137.7	2.1	.32	10	25.6	1592	13	30				
255	TC2JHC	34	0	340	148.5	0.5	.17	19.9	23	3	216	12	97			
256	TC2JHF	163	5	981	140	2	1	3	20	2	1170	13	64			
257	TC2JHF	130	9	844	162	8	1.0	.25	15	0	26.7	853	12	33		
258	1 PC2HF	65	2	652	170	0	1	45	22	18	0	23	7	978	11	17
	2 N	65	1	651	170	3	1.5	.22	18	0	23	8	980	11	30	
259	PC2NF	74	5	745	190	0	0.9	.32	14	4	27.8	7227	11	30		
271	1 TC2NF	76	3	763	198	6	1.6	.25	18		25.6	5332	12	10		
	2 JH	76	3	763	199	4	2.7	.25	18		26.4	5449	12	68		
272	TC2JHF	60	5	726	165	7	2.2	.26	22	8	20.5	692	12	67		
273	PC2JHF	80	1	641	165	3	1.6	.29	24	6	18.7	662	12	64		
274	1 TC2NF	80	1	641	186	6	0.9	.29	20	6	23.3	567	11	39		
	2 JH	80	1	641	184	4	1.2	.24	24	5	23.3	566	11	49		
275	TC2JHF	71	3	445	95	3	1.17	.19	12	3	21.6	130	12	62		
276	1 PC2JHF	66	0	585	107.5	1.20	.23	9	4		24	5	578	12	42	
	2 T	66	2	587	107.3	1.02	.20	10	8		24.0	567	15	11		
277	TC2JHF	90	0	648	133.4	1	.58	.12	15	4	21.9	884	14	7		
278	PC2JHF	90	3	770	133	8	2.86	.20	9	7	27.4	956	11	35		
279	PC2JHF	91	6	691	157	3	1.5				33	6	1040	10	1	
291	1 PC2HF	102	3	716	157	0	1	5	.2	9	3	31	8	430	9	3
	2 T	102	2	715	159	8	2.3	.27	10	7	30	8	407	13	2	
292	1 PC2HC	100	7	604	132	1	1	0	.26	9	1	26	1	481	9	1
	2	100	7	604	131.7	1.0			.15	13	7	18	8	348	8	3
301	1 TC3NC	83	6	501	130.4	3	0	.30	17	3	16	6	114	15	2	
	2 J, H	84.7	508	129	6	2.5	.30	20	4		17	2	120	13	7	
302	TC3JC	74	6	448	136	6	1.36				26	5	234	13	2	
303	TC3NF	76	0	608	178.	1	0	.30	22		25	8	785	11	7	
304	1 PC3JF	100	3	602	214.	2	0				35	6	1177	16	3	
	2 N	102.4	614	207.	2.1						29	7	1630	10	6	
311	TC3JHF	70	3	471	160.5	0	.84	.28	18	2	24	9	205	12	61	
312	TC3JHF	58	6	596	163.	.93	.23	21	5		21.9	802	12	4		
313	TC3JHF	66	7	592	170.	.88	.22	22	5		22	3	554	11	38	
314	1 TC3JHF	144	6	618	182	9	1.27	.25	18	3	26	3	538	12	3	
	2 P	144	2	617	180	8	1.24	.29	16	2	26	8	549	9	6	
315	PC3JHF	85	6	505	157.6	1.0					21	1	715	10	30	
316	1 TC3JHF	82	7	703	217	5	1.1				25	2	2976	11	75	
	2 P	82	7	703	212.7	1.1					24	1	2850	9	62	

For meaning of symbols see pages 602 to 606.

CORLISS TYPE.

TABLE 75 C.

m, lb	l <sub>2</sub>	Q <sub>H</sub>	h	Q	W	w	W <sub>R</sub>	E <sub>A</sub>	E <sub>R</sub>	Q <sub>M</sub>	No.
.012	151		-10.5	1055	154.0	-2.5	225.6	.146	.683	290	251.1
.009	144	-32.1	-7.8	1035	141.2	-1.9	224.9	.137	.578	311	2
.02	133	-13.2	-17.4	1055	124.4	-4.6	254.5	.118	.489	360	252.1
.02	133		-17.4	1069	131.5	-4.6	255.4	.123	.515	344	2
.01	111		-8.8	1096	156.4	-2.5	280.0	.143	.559	307	253
14.5°	128		+9.2	1102	188.6	+2.4	275.9	.171	.685	248	254
0	80	-32.0	0	1111	196.3	0	347.5	.177	.565	240	255
.007	111	-31.0	-6.1	1074	195.3	-1.8	296.2	.182	.659	233	256
.007	102	-26.4	-6.0	1091	206.4	-1.9	319.2	.189	.647	224	257
90.5°	115	-13.7	+57.3	1156	227.8	+19.6	325.7	.197	.700	215	258.1
72.7°	116		+46.3	1157	219.5	+15.1	319.7	.190	.687	223	2
9.4°	99		0	1137	214.0	+1.9	328.1	.188	.651	225	259
.007	118		-5.9	1106	210.3	-1.9	306.2	.190	.688	223	271.1
.007	138	-17.5	-5.9	1069	200.7	-1.7	279.1	.188	.720	220	2
.006	130	-18.6	-5.1	1072	201.0	-1.5	282.0	.188	.714	226	272
14.7°	118	-28.6	+9.5	1088	211.4	+3.0	295.4	.195	.716	218	273
41°	99		+26.6	1157	225.5	+9.5	344.6	.195	.654	218	274.1
41°	108	-36.5	+26.6	1110	221.5	+9.3	330.3	.200	.671	213	2
.01	107	-25	-8.9	1072	201.7	-2.5	276.5	.178	.730	238	281
129.3°	108	-20	+76.4	1164	204.9	+25.4	301.0	.176	.681	241	282.1
.01	103	-20	-8.8	1084	168.4	-2.6	290.8	.155	.579	273	2
.01	118	-22	-8.7	1072	172.3	-2.5	283.4	.161	.608	264	233
106.6°	140	-15	+65.2	1130	214.8	+19.3	274.5	.190	.782	223	284
199°	116	-20	+120.0	1208	250.0	+44.5	343.9	.207	.727	205	285
375°	116	0	+215.5	1324	266.2	+86.7	386.0	.201	.690	212	291.1
.01	132	-10	-8.6	1074	183.7	-2.4	276.4	.171	.665	248	2
395°	105	0	+221.	1336	279.7	+95.0	402.7	.206	.696	200	292.1
402°	105	0	+225.	1340	286.3	+97.6	405.1	.217	.707	190	2
.008	142		-7.0	1071	166.6	-1.8	249.4	.155	.668	273	301.1
.008	135	-45.1	-7.0	1033	182.8	-1.8	258.6	.177	.707	240	2
.0117	112	-27.2	-10.8	1073	192.8	-3.0	291.0	.179	.663	236	302
.01	102		-8.5	1117	216.6	-2.9	324.2	.194	.667	219	303
87.4°	126	-6.6	+58.0	1157	244.2	+20.0	325.4	.212	.752	201	304.1
91.5°	128		+60.5	1163	244.7	+20.2	321.0	.211	.763	202	2
0	96	-45	0	1084	201.9	0	328.8	.186	.614	228	311
.007	100	-30	-6.	1090	204.3	-1.9	322.9	.188	.633	226	312
.007	98	-30	-6.	1092	212.4	-1.9	328.2	.195	.647	217	313
0	110	-25	0	1093	205.9	0	317.5	.188	.649	225	314.1
230°	109	-20	+140.	1238	254.7	+54.4	372.3	.206	.683	206	2
140.5°	102	-22	+86.5	1187	233.4	+32.2	351.3	.197	.665	216	315
0	105	-20	0	1107	216.5	0	335.5	.196	.645	217	316.1
213°	105	-15	+132.	1244	264.6	+54.0	388.3	.213	.682	190	2

es mean assumed values.

TABLE 75 D.

PUMPING AND

No.	DIAMETERS			×	STROKE	RATIO	CLEARANCES	TYPE
WORTHINGTON TYPE PUMPING ENGINES								
331	30,	60	×	36	4.05			Hor. duplex Knowl's
332	12, 18,	29	×	18	5.83			Hor. duplex Worth'n
336	18,	36	×	26	4.05			Hor. high-duty
337	33,	66	×	37.5	4.05			Hor. high-duty
COMPOUND FLY-WHEEL PUMPING ENGINES								
341	21,	42	×	36	4.04	.036-.025		Hor. 2-ck. Holly
342	15,	30	×	30	4.09	.040-.037		Hor. 2-ck. Corliss
343	27,	54	×	120	4.02	.016-.015		Vertical Leavitt
344	34,	62	×	48	3.34	.012-.011		Vert. Allis-Chalmers
TRIPLE-EXPANSION PUMPING ENGINES								
351	24,	38,	54	×	36	5.06	.025-.020	Hor. 3-ck. Corliss
352	21,	37,	55	×	48	6.95	.018-.021	Hor. 3-ck. Nordberg
353	28,	48,	74	×	60	7.11	.014-.008	Vert. 3-ck. Reynolds
354	14.2, 23.6,	34.4	×	39.4	6.19	.055-.060		Hor. 2-ck. Sulzer
355	13.7, 24.4,	39	×	72	8.11			Vert. 3-ck. Leavitt
356	20,	52,	80	×	60	7.66	.017-.025	Vert. 3-ck. Snow
357	22,	41.5,	62	×	60	7.95	.010-.005	Vert. 3-ck. Holly
358	30,	56,	87	×	66	8.45	.014-.005	Vert. 3-ck. Allis-Chal.
ENGINES WITH REGENERATIVE FEED-HEATERS (NORDBERG)								
365	19.5, 29, 49.5, 57.5	×	42	8.75	.013-.004			Vert. 2-ck. Pump. Eng.
366	14.5, 22, 38, 54	×	48	14.19	.060-.035			Hor. 4-ck. Air-comp'r
COMPOUND MARINE ENGINES								
371	16,	25	×	24	2.44	.049-.041		Vert. tandem
372	27.4,	50.3	×	33	3.40	.076-.050		Vert. 2-crank
373	30,	57	×	36	3.65	.094-.062		Vert. 2-crank
374	50.1,	97.1	×	72	3.78	.15-.12		Inclined 2-crank
MULTIPLE-EXPANSION MARINE ENGINES								
381	18.5,	27,	42	×	24	5.18	.160-.113	One, vert. 3-crank
382	29.4,	44,	70	×	48	5.70	.124-.080	One, vert. 3-crank
383	22,	34,	57	×	39	6.75		One, vert. 3-crank
384	33,	49,	74	×	39	5.02		Two, vert. 3-crank
385	34,	55.5,	64-64	×	48	7.12		Two, vert. 4-crank
386	29,	41.5,	59, 84	×	54	8.45		Two, vert. 4-crank

MARINE ENGINES.

TABLE 75 D.

SERVICE	LOCATION	YEAR	REPORTED BY	REFERENCE	No.
Reservoir	Birm'gham, Ala.	1893	G. H. Barrus	E. R., Jan. 13, '94-109	331
Reservoir	St. Albans, Eng.	1899	E. H. Foster	A. S. M. E., 21-788	332
Experi'tal	Brooklyn	1885	J. G. Mair	q Eng. 1886, II. 340	336
Oil pipe-line	Pt. Jarvis, N.Y.	1891	J. E. Denton	A. S. M. E., 12-975	337
Reservoir	S. Bethlehem, Pa.	1900	R. C. H. Heck		341
City water	Pawtucket, R.I.	1891	Wm. Kent	A. S. M. E., 13-176	342
Reservoir	Louisville	1894	F. W. Dean	A. S. M. E., 16-169	343
Reservoir	Omaha	1900	R. W. Hunt & Co.	E. R. Ju. 16, '00-574	344
Oil pipe-line	Laketon, Ind.	1893	J. E. Denton	A. S. M. E., 14-1341	351
Into mains	Grand Rapids	1896	M. E. Cooley—	A. S. M. E., 21-1019	352
City water	Milwaukee	1893	R. C. Carpenter	E. R., Dec. 2, '93-5	353
Lift of 1300 ft.	St. Gallen, Sw.	1897	A. Stodola	Z. V. D. I. '98, 265	354
City water	Boston	1895	E. F. Miller	Mass. In. Tech. Rep.	355
Into mains	Indianapolis	1898	W. F. M. Goss	A. S. M. E., 21-793	356
City water	Boston	1901	D. Brackett—	E. R., N. 16, '01-474	357
City water	Boston	1900	D. Brackett—	E. R., O. 13, '00-345	358
City water	Pittsburg	1899	R. C. Carpenter	E. R., Ap. 29, '99-495	365
Mine mach'y	Painesdale, M'h.	1906	O. P. Hood	A. S. M. E., 28-	366
Steamer	Bache	1874	C. E. Emery	Eng. 1875, I. 14	371
Steamer	Fusi Yama	1888	A. B. W. Kenne- dy, Committee	q Eng. 1890, I. 577	372
Steamer	Colchester	1889		q Eng. 1890, I. 605	373
Paddle str.	Ville de Douvres	1891		q Eng. 1892, I. 586	374
Ferryboat	Bergen	1888	E. A. Stevens—	A. S. M. E., 11-372	381
Steamer	Meteor	1888	A. B. W. Kenne- dy, C. In. M. E.	q Eng. 1889, I. 527	382
Steamer	Iona	1890		q Eng. 1891, I. 568	383
Cruiser	Minerva	1896	Adm. Com. R. N.	q Eng. 1902, I. 326	384
Cruiser	Argonaut	1898	Sir J. Durston	q Eng. 1899, I. 432	385
Steamer	Saxonia	1900	Adm. Com. R. N.	q Eng. 1902, I. 326	386

TABLE 75 D.

PUMPING AND

No.	CONDITION	N	V	P <sub>1A</sub>	P <sub>2A</sub>	e	r	P <sub>m</sub>	H	S <sub>H</sub>	J
331	TC2JHF	16.1	98	105.6	2.3	1.00	4.0	29.2	477	29.23	
332.1	PC3JHF	33.7	103	104.2	2.4	1.00	5.7	25.7	105	16.59	67
2	N	33.0	102	103.7	2.4	1.00	5.7	25.6	105	16.94	---
336	PC2JHF	45.0	195	74.0	1.5	.45	8.2	20.9	119	17.17	15
337	TC2JHM	20.1	126	103.6	2.2	.33	13.8	17.5	457	16.40	10
341	TC2JF	20.2	121	104.7	2.0	.42	9.1	27.7	281	15.63	12
342.1	PC2JF	48.2	241	138.1	1.05			27.7	142	13.84	67
2	N	48.0	240	138.3	1.4			27.6	141	14.26	---
343	TC2JHF	18.6	372	151.6	1.0	.20	20.2	25.0	643	12.22	107
344	TC2JHC	22.7	181	118.3	1.5	.15	20.5	20.8	357	12.35	114
351.1	TC3JHF	27.8	167	165.4	1.78	.22	20.8	27.9	323	13.83	17
2	N	27.7	166	165.6	1.84	.31	15.4	28.6	328	14.09	---
3	J <sub>1</sub>	27.8	167	127.7	1.83	.36	13.4	27.9	322	14.33	113
352.1	TC3JHF	39.9	319	138.8	1.72	.31	22.1	22.2	510	12.74	118
2		30.1	241	139.4	1.79	.28	23.9	21.8	376	13.33	122
3		19.1	158	140.3	1.76	.25	27.2	20.9	229	13.90	125
353	TC3JHF	20.3	203	135.9	1.2	.35	19.6	21.7	574	11.80	6
354.1	PC3JF	60.3	395	162.1	1.4			18.9	207	11.70	124
2		30.6	201	165.2	1.3			18.1	101	12.78	128
355	TC3JHC	50.6	607	190.7	1.45	.38	20.6	26.5	576	11.22	129
356	TC3JHC	21.2	212	168.9	1.6	.33	22.6	24.4	783	11.50	130
357	TC3JHC	24.9	249	165.6	1.4			20.5	465	11.09	131
358	TC3JHC	17.7	195	201.9	0.83			22.8	802	10.48	132
365	TC4JHC	36.5	255	214.1	1.15			35.4	712	12.42	
366	TC4JHC	56.9	455	256.8	1.25			31.3	990	11.92	
371.1	TC2NF	49.3	197	95.0	2.6	.41	5.6	29.8	86	23.21	---
2	J <sub>1</sub>	54.7	219	94.7	2.2	.39	6.9	32.7	105	21.25	133
372	TC2NF	55.6	306	71.5	2.3	.64	5.6	19.9	371	21.17	---
373	TC2NF	86.6	532	95.2	2.5	.62	5.4	24.8	1980	21.73	---
374	TC2NM	36.8	442	120.5	4.7	.63	5.4	30.2	2977	19.41	---
381	TC3NF	144	576	129	1.4	.63		27.5	665	18.3	---
382	TC3JF	71.8	574	153	2.73	.50		29.9	1994	14.98	
383	TC3JF	61.1	398	179.6	0.70	.37		21.1	645	13.35	134
384.1	TC3JC	83.5	642	95	1.3			15.2	2142	15.90	135
2		109.5	711	148	1.75			26.9	4963	14.13	
3		128.6	835	151	1.8			37.4	8132	16.30	---
385.1	TC3NC	76.4	611	83	1.9	.73	8.3	16.2	3867	17.72	---
2		75.1	601	184	2.1	.27	15.7	16.1	3762	16.26	---
3		116.5	932	189	1.9	.73	8.3	37.9	13766	15.98	---
4		115.3	922	245	1.7	.53	10.4	38.3	13788	15.44	---
5		127.7	1022	259	2.1	.71	8.5	47.2	18781	15.75	---
386	TC4NC	77.8	700	207	2.3			38.8	9099	13.47	---

For meaning of symbols see pages 602 to 606.

MARINE ENGINES.

TABLE 75 D.

<i>m, lb</i>	<i>h</i>	<i>q<sub>H</sub></i>	<i>h</i>	<i>Q</i>	<i>W</i>	<i>w</i>	<i>W<sub>R</sub></i>	<i>E<sub>A</sub></i>	<i>E<sub>R</sub></i>	<i>Q<sub>M</sub></i>	No.
020	132	<i>-25</i>	-17.6	1041	87.0	-4.5	246.6	.084	.353	508	331
43°	133	-16.1	+83.2	1149	153.4	+26.2	274.0	.134	.561	317	332.1
54°	133		+89.3	1171	150.3	+28.7	276.2	.128	.544	330	2
2.5°	116	-23.9	+28.1	1096	148.3	+7.5	257.4	.135	.577	313	336
01	130	-32.6	-8.8	1042	155.2	-2.2	249.8	.149	.622	284	337
01	126	-26.4	-8.8	1053	162.8	-2.3	255.5	.154	.635	276	341
3.5°	104	-14.3	+2.2	1105	184.0	+0.9	307.8	.167	.598	255	342.1
0.9°	114		+6.9	1115	178.5	+2.6	296.9	.156	.601	272	2
0055	102	-43.7	-4.8	1073	208.3	-1.5	315.2	.194	.661	218	343
01	116	-25.9	-8.7	1067	206.1	-2.5	278.2	.194	.740	219	344
0217	122	-44.2	-18.6	1041	183.5	-5.5	288.5	.176	.636	241	351.1
020	123		-17.9	1084	180.9	-5.0	287.5	.167	.629	254	2
0225	123	-33.6	-19.5	1043	177.7	-5.4	270.3	.170	.657	249	3
0	121	-26.4	0	1074	199.8	0	284.3	.186	.703	228	352.1
0	122	-30.0	0	1069	191.0	0	282.5	.179	.677	237	2
0	122	-37.0	0	1063	183.0	0	283.9	.172	.645	247	3
0105	108	-19.9	-9.1	1084	215.7	-2.7	298.0	.199	.723	213	353
269°	114	-32.0	+159	1239	219.2	+63.0	367.7	.177	.596	240	354.1
262°	111	-33.0	+155	1237	199.2	+61.9	371.6	.161	.537	264	2
0	115	-36.4	0	1078	226.8	0	313.7	.211	.723	202	355
010	118	-13.7	-8.5	1086	221.3	-2.6	298.1	.204	.743	208	356
007	114	-27.5	-6.0	1075	229.5	-1.8	304.2	.214	.755	199	357
0137	96	-36.5	-11.5	1087	242.8	-3.9	340.5	.224	.714	190	358
0126	106.7	-206.0	-10.6	909	204.9		.326	.226	.691	188	365
0574	109.5	-228.1	-47.8	851	213.5		.334	.251	.751	169	366
02	136		-17.7	1059	109.7	-4.2	233.1	.104	.472	408	371.1
02	130	-10.2	-17.7	1055	119.7	-4.4	241.7	.114	.496	373	2
02	132		-18.0	1057	120.2	-4.1	221.0	.114	.544	373	372
02	135		-17.7	1060	117.1	-4.3	235.4	.110	.498	384	373
02	160		-17.5	1041	131.1	-4.0	217.1	.126	.605	336	374
008	114		-7.0	1099	139.0	-2.0	287.5	.127	.483	336	381
015	138	-15	-12.9	1058	170.0	-3.5	263.4	.161	.645	264	382
015	90	-12.0	-12.7	1115	190.7	-4.3	340.0	.171	.561	248	383
0	111	-3.8	0	1108	160.1	0	273.5	.145	.587	292	384.1
01	121	-8.6	-8.6	1084	180.2	-2.5	285.1	.166	.633	255	2
01	122	-5.3	-8.6	1087	156.2	-2.5	285.0	.144	.548	295	3
0	124		0	1085	143.7	0	245.1	.133	.587	320	385.1
0	128		0	1100	156.5	0	292.9	.142	.534	298	2
0	124		0	1104	159.2	0	299.7	.144	.532	294	3
015	120		-12.4	1103	164.8	-4.0	318.7	.149	.517	284	4
005	128		-4.1	1105	161.6	-1.3	314.5	.146	.514	290	5
015	132		-12.6	1087	189.0	-3.8	293.2	.174	.645	244	386

Italic figures mean assumed values.

TABLE 75 E. STEAM-TURBINES.

No.	Size	Type	Location	Year	Reported by	Refs.
SINGLE-STAGE ONE-IMPULSE TURBINES.						
401	30 H.P.	De Laval	Dresden	1901	E. Lewicki	Z. V. D. M.
402	50 H.P.	De Laval	Glasgow	1905	T. B. Morley	Eng. 190
403	100 H.P.	De Laval	Lods	1901		Z. V. D. M.
404	300 H.P.	De Laval	Trenton	1902	Dean & Main	E. R. A. 2
409	2000 H.P.	Riedler-Stumpf	Berlin	1902	A. Riedler	g. Stod. II. 1
MULTIPLE-STAGE SINGLE-IMPULSE TURBINES.						
411	500 H.P.	Rateau	Paris	1903	A. Stodola	Stod. II. 1
412	1000 Kw.	Rateau	Zurich	1903	Oerlikon Works	Stod. II. 1
413	300 H.P.	Rateau L.P.		1902	Sauvage & Picou	Stod. II. 1
414	700 H.P.	Rateau L.P.	Hallside W'ks	1903		Eng. 190
421	500 H.P.	Zoelly	Zurich	1904	A. Stodola	Stod. II. 1
MULTIPLE-STAGE MULTIPLE-IMPULSE TURBINES.						
431	50 H.P.	Elektra		1905	Prof. Gutermuth	Stod. II. 1
441	500 Kw.	Curtis	Newport, R. I.	1904	G. H. Barrus	Report
442	500 Kw.	Curtis	Cork, Ire.	1904	C. H. Merts	Eng. 144
443	500 Kw.	Curtis	Revere, Mass.	1906	W. H. Trask, Jr.	* Bulletin
444	2000 Kw.	Curtis	Schenectady	1905	Sargent & Ferguson	Report
445	5000 Kw.	Curtis	Schenectady	1906	W. L. R. Emmet	Gen. Eng.
PARSONS TURBINES.						
451	400 Kw.	Westinghouse	Pittsburg	1903	Dean & Main (55)	F. Holtz
452	1250 Kw.	Westinghouse	Pittsburg	1903	J. Kennedy (41)	A. S. M. E.
453	3000 Kw.	Brown-Boveri	Frankfort-a-M.	1904	—Singer	g. Z. V. D. M.
MIXED-TYPE TURBINES.						
461	50 H.P.	Union	Essen	1905		† St. & H.
466	500 Kw.	H.-P.-S. ††	Munich	1906	M. Schroter	Eng. 140

\* "The Vertical Steam Turbine." Paper by C. B. Barleigh, Nat. Assoc. Cotton Mfrs. 1902.  
† Stevens & Hobart, "Steam Turbine Engineering."  
†† H.-P.-S. = Helms-Pfenninger-Sankey.

ADDITIONAL DATA FOR TABLE 75 F, SHOWING PARTICULARS AS TO ENGINES NOT INCLUDED IN PRECEDING TABLES (ITALIC NUMBERS).

229	Scott Central-valve vert. compound				Eng. 140
230	Two Reavell engines, vert. compound				Eng. 140
245	23,	48 × 48	} McIntosh & Seymour } vert. two-crank	L. S. Marks	A. S. P.
246	29,	60 × 56			
261	16,	30 × 42	Hor. tand. Corliss	E. J. Willis	g. E. R. A.
262	20,	38 × 48	Hor. cross-comp. Corliss	Stone & Webster	E. R. A.
263	26,	50 × 48	Hor. cross-comp. Corliss	G. H. Barrus	E. R. A.



TABLE 75 F. ENGINE AND GENERATOR EFFICIENCIES.

No.	COND.	SIZE *	N	P <sub>1A</sub>	t <sub>b</sub>	H	K	K/H	E <sub>C</sub>	S <sub>H</sub>	S <sub>K</sub>
116	TA2	16 ×14	265	115	—	108	67	.627	.841	23.1	36.8
114	TA2	20 ×13	271	181	—	188	161	.638	.856	21.1	33.1
129.1	TA2	16 ×8	408	175	—	84	48	.576	.772	19.2	33.0
2			412	175	—	43	24	.563	.755	21.5	38.2
3	C		412	172	—	94	58	.610	.818	16.8	27.6
4			412	174	—	43	24	.568	.761	17.3	30.4
130.1	TA2	150 Kw.	346	150	—	246	150	.612	.82	19.1	31.3
2	TC2	100 Kw.	446	150	—	160	100	.626	.84	15.5	24.7
124	PA2		302	169	107°	600	358	.597	.800	15.9	26.8
125.1	TC2			167	—	360	226	.627	.841	16.9	27.0
2	P			173	150°	360	227	.629	.843	13.4	21.3
127	PC3			180	112°	1298	775	.602	.807	11.1	18.3
136	TC2	40½ ×27	150	167	—	502	323	.664	.864	12.7	19.7
145	PC2	48 ×48	102	164	9°	1047	674	.643	.863	13.6	21.3
146.1	PC2	60 ×56	98	169	80°	2267	1540	.678	.910	12.7	18.7
2			98	163	98°	2202	1495	.678	.910	11.6	17.1
152	TC2	22 ×42	114	126	—	185	122	.643	.864	19.9	30.9
161	TC2	30 ×42	100	127	—	373	219	.587	.787	13.8	23.5
162	TC2	38 ×48	107	127	—	709	469	.661	.886	15.3	23.1
163	TC2	50 ×48	78	151	—	1031	690	.668	.896	13.2	19.8
171.3	TC2	76.6 ×60	76	209	—	1482	867	.606	.814	15.9	26.2
4			76	198	—	5274	3740	.708	.950	12.2	17.2
5			76	187	—	6326	4483	.708	.948	12.4	17.6
159.2	PC2	86 ×60	75	190	6°	7365	5079	.690	.924	12.0	17.3
186.1	TC2	22 ×33.5		200	—	308	190	.618	.829	13.6	22.0
2				192	—	165	99	.598	.802	11.8	19.1
3	P			200	225°	310	190	.614	.823	10.9	17.7
4				200	225°	165	99	.598	.802	9.7	16.2
193.1	PC2	24 ×28	140	155	390°	312	190	.610	.817	9.4	15.4
2			140	155	390°	239	143	.600	.804	9.0	15.0
3			140	155	390°	175	97	.554	.743	9.5	17.2
104.1	PC3	48 ×36	100	214	87°	1177	719	.610	.817	10.4	17.1
3			100	213	84°	850	503	.590	.792	9.9	16.8

For meaning of symbols see pages 602 to 606.  
\* Under "Size" are here given the diameter of the L.P. cylinder and the stroke.

SUPPLEMENT TO TABLE 75 F—Continued.

186	12.8, 22.0×33.5 Van den Kerchove (Fig. 506) horizontal tandem 250 H.P. engine, Ghent, 1902 M. Schroeter ENG'R, 1903 I. 192
193	15, 24×28 horiz. tandem, lift valves, positive gear, Schmidt separate superheater. Brantham, 1902 J. A. Ewing ENG'R, 1903 I. 46

TABLE 75 E.

TESTS OF

No.	CONDITION	N	P <sub>1A</sub>	P <sub>2A</sub>	K	B	E <sub>2</sub>	E <sub>3</sub>	K/H	S <sub>2</sub>
401	1 TABC	2014	99	1 14 5		43 5		94		39 5
	2 P	2354	99	1 14 45		25 0		92		33 5
	3 P	2096	99	3 14 65		51 2		95		25 5
402	TCEF	1635	187	1 59		48 0		95		21 5
403	TCBF	1076	186	2 2 28		242 1		95		17 30
404	1 TCEF	747	221	1 1 67		333 0		95		15 7
	2	751	216	2 0 84		118 9		92		16 7
	3 P	750	221	8 1 47		352 0		95		15 4
409	1 PCEC	3000	203	2 14	1365	1928	.95	.97	688	13 48
	2 B	3800	185	1 22	1340	1891	.95	.97	688	12 48
411	1 TCEC	2184	176	1 1 29	107 5	171 6	.84	.90	564	30 4
	2 P	2101	168	5 1 64	366 0	530 8	.924	.95	655	22 4
	3 P	2360	223	7 2 15	462 9	664 9	.933	.96	669	22 4
412	1 TCEC	1500	179	2 2 43	1024		.95	.97	688	21 48
	2	1500	186	3 1 11	194		.85	.90	571	31 7
413	PAEC	1598	14 7	2 79	232 5		.93	.96	666	34 4
414	TAEC	1500	14 7	1 0	450		.94	.96	673	36 4
421	1 TCEC	2967	158	8 1 03	387 7		.94	.96	673	21 18
	2	2995	157	1 0 74	80 1		.84	.90	564	33 4
	3 P	2970	160	1 0 99	390 4		.94	.96	673	19 4
431	1 PCEC	3524	145	8 2 43		44 4		.94		27 4
	2	3181	143	9 1 78		58 5		.95		30 4
	3 A	3331	146	7 14 7		38 8		.94		32 4
441	1 TCEC	1815	165	1 0	520 4		.93	.96	660	19 7
	2 P	1815	165	1 0	514 7		.93	.96	660	15 7
442	PCTC	1820	168	1 5	512		.94	.96	660	20 7
443	TCFC	1820	167	0 6	521 9		.93	.96	660	18 7
444	1 PCTC	900	181	0 73	2024		.945	.96	676	15 7
	2	900	170	2 0 71	555		.90	.92	631	18 7
445	PCTC	750	190	1 0	4800		.955	.97	691	14 7
451	1 TCBC	3549	160	6 1 46		593 5		.96		14 7
	2	3545	169	1 0 90		593 5		.96		13 7
	3 P	3544	164	9 0 98		593		.96		12 7
	4	3543	168	2 0 93		592		.96		11 7
452	1 CCEC	1196	161	7 1 42	1322		.95	.97	688	19 4
	2 P	1201	160	9 1 42	1294		.95	.97	688	18 4
	3 T	1197	160	7 0 94	1364		.95	.97	688	18 4
	4 P	1199	160	6 0 93	1274		.95	.97	688	17 4
453	1 PCEC		199	1 42	3000		.95	.97	688	14 4
	2		142	1 42	3000		.95	.97		16 4
461	TCBC	3542	157	3 1 4				.95		20 4
466	TCFC	2518	191	0 48	504			.96	680	17 4

For meaning of symbols, see pages 603 to 606

## STEAM-TURBINES.

TABLE 75 E.

$\alpha$	$m, t_s$	$t_2$	$h$	$Q$	$W$	$w$	$W_R$	$E_A$	$E_R$	$Q_M$	No.
16	0	211	0	1002	68.5	0	139.2	.068	.492	620	401.1
14	327°	211	+181	1183	81.8	+51.3	190.5	.069	.430	615	2
40	604°	212	+322	1323	104.3	+97.7	202.0	.079	.517	538	3
80	.007	118	-5.9	1105	122.4	-1.8	306.0	.111	.400	382	402
34	30.6°	131	+20	1117	155.8	+6.4	295.8	.139	.527	304	403
75	.0215	120	-18.0	1095	172.7	-5.8	311.0	.158	.555	269	404.1
34	.0215	96	-18.0	1118	166.0	-6.2	341.7	.149	.487	285	2
24	84°	115	+55.8	1174	192.2	+19.3	342.5	.164	.562	259	3
45	179°	129	+113	1215	189.4	+40.4	342.0	.156	.554	272	409.1
98	197°	109	+121	1241	212.4	+46.3	366.2	.171	.580	248	2
16	5.2°	111	+3.4	1120	148.3	+0.9	315.1	.124	.472	359	411.1
80	20.0°	119	+13.0	1120	172.0	+3.9	303.3	.143	.567	310	2
77	23.6°	129	+16.2	1119	172.3	+5.3	307.7	.144	.561	307	3
11	.0052	133	-4.3	1090	168.5	-1.3	282.3	.155	.597	274	412.1
24	.0153	105	-13.0	1110	139.6	-4.2	321.0	.126	.435	338	2
62	84.5°	139	+40.5	1080	95.6	+6.4	115.8	.089	.825	480	413
65	0	102	0	1077	103.2	0	168.3	.096	.615	442	414
46	7.0°	103	+4.5	1126	176.2	+1.2	320.6	.157	.549	271	421.1
66	3.1°	92	+2.0	1134	136.4	+0.5	334.7	.120	.408	353	2
33	76.5°	102	+48.2	1171	191.0	+18.4	338.7	.163	.563	260	3
73	196°	134	+116.8	1206	99.0	+37.6	307.3	.082	.322	517	431.1
15	105°	122	+64.5	1165	132.9	+22.2	307.0	.114	.433	372	2
58	104°	212	+64.0	1074	83.3	+14.2	181.0	.078	.460	548	3
18	0	102	0	1123	193.2	0	322.1	.172	.600	246	441.1
59	290°	102	+176	1300	240.3	+73.8	395.9	.185	.608	229	2
66	104°	116	+65	1176	186.5	+22.1	325.8	.159	.573	267	442
48	.012	86	-10.2	1130	203.8	-3.3	343.4	.180	.595	235	443
15	207°	92	+126	1262	250.7	+51.4	394.5	.199	.635	214	444.1
40	204°	91	+125	1261	223.3	+50.7	391.0	.177	.573	239	2
34	150°	102	+89.5	1216	246.2	+31.2	362.5	.203	.680	209	445
78	0	115	0	1112	184.8	0	305.6	.166	.604	256	451.1
35	.001	102	-0.9	1123	190.7	-0.3	323.9	.170	.588	250	2
.00	104°	101	+65.3	1189	212.1	+23.8	346.8	.179	.612	238	3
.00	180°	100	+110	1237	231.3	+42.4	369.2	.187	.627	227	4
37	0	114	0	1111	190.4	0	303.9	.171	.626	248	452.1
70	76°	114	+47.8	1159	200.4	+16.2	319.7	.173	.647	245	2
92	0	100	0	1125	197.0	0	323.4	.175	.610	242	3
12	78°	100	+49	1174	210.0	+16.7	340.3	.179	.617	237	4
83	190°	114	+119	1235	258.9	+45.7	363.3	.210	.713	202	453.1
57	90°	114	+55.5	1163	220.0	+18.7	314.0	.189	.700	224	1
57	117°	114	+72	1193	130.1	+25.2	327.9	.109	.397	388	461
67	73°	79	+137	1288	218.0	+52.5	423.9	.170	.514	250	466

Values in parentheses mean assumed values.

TABLE 75 G. MECHANICAL EFFICIENCY OF ENGINES.

No.	Cond.	Size	N	P <sub>1</sub> A	H	B, D	Diff. E <sub>m</sub>	No.	Cond.	Size	N	P <sub>1</sub> A	H	D, P	Diff. E <sub>m</sub>		
Power Engines.								Locomotives—Continued.									
1151	TA1	7½×10	172	135	13.9	11.4	2.5	819	202.1	TA2	23.7×25.2	100	229	631.4	509.6	121.8	807
1152	TA1	8½×15	136	140	23.4	17.0	6.4	724	2			240	230	1257.9	1136.2	121.7	905
211	TA2	9×11	173	140	21.1	17.3	3.8	818	203.1	TA2	25×26	160	237	1296	1144	152	884
212	TA2	9½×10	160	265	18.6	17.6	1.0	947	2			240	232	1529	1199	330	784
213	TA2	9×15	149	170	24.0	22.0	2.0	877	204.1	PA2	22.1×23.2	80	216	481	447	34	930
172	PA1	7.1×11	200	143	22.8	18.6	4.2	818	2			160	214	742	653	89	881
221	PC2	27.3×19.8	117	183	73.3	62.0	11.3	847	3			240	208	744	613	131	623
PUMPING-ENGINES.																	
112.3	TC1	9×36	84	113	73.1	70.7	2.4	967	331	TC2	60×36	16	106	477.0	454.3	22.7	963
4			87	115	30.3	27.6	2.7	912	332	PC3	29×18	33	104	105.1	96.5	8.6	923
5			89	114	9.5	7.5	2.0	788	336	PC2	36×26	45	74	119.2	109.3	9.9	917
251.3	TC2	16×36	84	121	98.6	92.1	6.5	934	337	TC2	66×37.5	20	104	456.7	439.9	16.8	964
4			85	121	67.1	60.7	6.4	906	343	TC2	54×120	19	152	643.4	599.1	44.3	931
5			86	124	35.7	29.4	6.3	824	344	TC2	62×48	23	118	356.8	308.9	47.9	869
6			88	123	13.8	8.1	5.7	587	351	TC3	54×36	28	165	322.6	301.0	21.6	933
301.3	TC3	24×36	83	132	141.0	122.5	18.5	870	352	TC3	55×48	40	139	509.8	477.0	32.8	937
4			85	129	89.2	72.3	16.9	811	2			30	139	376.4	300.1	16.3	932
5			88	130	24.6	9.6	14.9	392	3			19	140	228.6	218.6	10.0	956
302	TC3	30×36	75	137	234.3	223.9	10.4	956	353	TC3	74×60	20	136	573.9	517.7	56.2	902
Locomotives.																	
181	TA1	22×28	40	218	451.5	373.5	81.0	802	354	PC3	34.4×39.4	60	162	206.8	167.8	39.0	810
2			80	215	779.3	629.8	149.5	808	2			31	105	101.0	83.8	17.2	830
3			120	210	100.5	81.8	18.7	813	355	TC3	39×72	51	191	575.7	515.1	60.6	896
4			160	196	90.0	74.3	15.7	760	356	TC3	80×90	21	169	782.9	734.5	48.4	938
5			101	187	80.3	50.0	30.3	742	357	TC3	92×90	25	106	404.5	418.4	13.9	1015
601	TA2	36×42	11	224	177.3	140.5	36.8	803	358	TC3	87×90	18	202	201.6	747.4	54.4	913
2									359	TC3	87×90	11	213	219.5	200.1	19.4	917

[From page 607]

The most extensive collection of data as to lift-valve engines (with a few of other types) is that of "One Hundred Tests" of engines built at Augsburg, by J. Krumper, in Z. V. D. I. for 1905, of which a number are used in the tables.

The two engines which are most markedly of American development are the large compound Corliss and the large triple-expansion pumping-engine. The use of the higher grades of expansion in stationary power engines belongs mostly to European practice.

After the early experiments of Isherwood and Emery, the comparatively small amount of data that is available as to the steam-economy of marine engines comes mostly from English sources.

As to the quality of these data, it must be understood that the results got by special tests are likely to be quite a little better than the average working of engines of the class. Tests are generally made when the engine is comparatively new and in good condition, with a minimum of leakage; further, poor results have a tendency to be suppressed rather than to be made public. Occasionally reports get into print which are too good to be true; a few such have been included in the tables, intentionally, and a few remarks in regard to them will be found in § 76 (n).

### § 76. Observations, Results, and Calculations.

(a) GENERAL CONDITIONS.—We shall now take up the matter presented in the numerical portion of Tables A to D, noting the character and variation of the leading data, reviewing the methods of calculating the thermal performance, and briefly summarizing the results shown—thus preparing the way for the discussion of certain important influences upon the performance of both engine and turbine. The quantities got by observation are given on the left-hand page (with one column of the second page), and are the first to be considered.

*Speed.*—The data under  $N$  and  $V$  (R.P.M. and piston speed) add a large number of actual examples to the summary presented in § 41 (u). In Table A we see the variation from 9 R.P.M. and 160 ft. in an old-fashioned marine engine (No. 102) to 100 R.P.M. and 800 ft. in a quick-running Corliss (No. 115). For contem-

porary practice, the widest contrast is shown in Table D, between pumping and marine engines. The abnormally high piston-speed in No. 355 is made possible by a mechanism which gives the pump a shorter stroke than that of the engine (4 ft. instead of 6 ft.). Running over these columns in the several tables we find that the general statements in § 41 (u) are very well supported.

Speed exerts quite an influence upon the performance of steam in the engine. On the mechanical side, or that of steam-movement, low speed is favorable; the high-speed engine must have large ports (which means large waste spaces), and even it is likely to show excessive pressure-losses. In this connection linear velocity is the directly determinant influence: although we must note that, with steam-passages of a proper area with reference to the size of piston and its mean speed, the short-stroke engine will have a relatively much larger clearance-volume than will a long-stroke machine of smaller R.P.M.—that is, if the valves can be placed at the ends of the cylinder in both cases. Consider thermal losses, high speed becomes desirable, in the form of high R.P.M. or the diminution of the actual time occupied by the cycle of operations within the cylinder.

Speed is, of course, a prime factor in the capital cost of a power plant; upon it depends very largely the rate of work that can be got out of a given size and weight (or, approximately, cost) of engine. In many cases, of course, the designed speed of the engine is fixed by the service for which it is intended.

(b) PRESSURES.—The limiting pressures— $p_1$ , from which steam is admitted, and  $p_2$  to which it is exhausted—determine the probable magnitude of the energy-transformation process; they are reduced to absolute values, so as to eliminate the variable atmospheric pressure and to put them into shape for use in subsequent calculations.

A properly full report of an engine-test should give steam pressure at boiler, at engine-throttle, perhaps in steam-chest or exhaust-pipe, and in condenser, besides the highest admission pressure and the mean and lowest exhaust-pressures from the indicator diagrams. Actual tests covering a wide range of conditions vary greatly as to the fullness

do give. In the tables it is intended that all the pressure-losses from the steam as supplied at the inlet point and all back-pressure above a nearby exhaust outlet shall be included in the working of the engine.

After an inspection of the boiler-pressure column, we may summarize what is shown as to various lines of practice about as follows now giving gage-pressure rather than absolute:

Going back to 1860, we see simple marine engines on steam at 25 lbs. Simple power engines of the usual types work on 70 to 90 lbs., compound engines on 110 to 150, triples and high-ratio compounds on 120 to 200—the higher values belonging to the more recent installations. The locomotive carries 180 to 220 lbs. In modern ships with water-tube boilers it is quite usual to make steam at 300 lbs., then reduce it to 250 lbs. at the engine, with the idea of drying the steam by throttling-action—although in the adoption of this scheme there does not seem to be a very clear appreciation of the smallness of the thermal action involved in this pressure-change, the difference in total heat being only 5 B.T.U.

In regard to pressure-losses, it may be stated as a rough generalization that for well-arranged engines which govern by cut-off and run on "full throttle", the drop from boiler to cylinder will range from 3 to 7 per cent., becoming even greater with early cut-off and more wire-drawing by the engine-valve. In non-condensing engines with free exhaust, the back-pressure above atmosphere is likely to be from 0.5 to 1.5 lbs., and is often a good deal higher. With a condenser showing an absolute pressure of 1.5 to 2.0 lbs. (26 to 27 ins. of vacuum), the additional back-pressure in the cylinder will usually be from 0.5 to 1.0 lb. In the turbine, where high vacuum can be much more effectively utilized, this pressure-difference is diminished by the employment of very large and short exhaust-connections.

The greatest back-pressure is found in the locomotive, due partly to rather small passages, very largely to the choke of the exhaust-nozzle. To get a working basis for the locomotive tests which would make this engine properly comparable with the others, an arbitrary rule was adopted by the writer, as follows:

If  $p_b$  is the average back-pressure above atmosphere in the

cylinder, measured from the diagrams, then the relation between  $p_b$  and the exhaust-pressure  $p_{2A}$  properly chargeable to the engine alone, as distinguished from the blast-apparatus, was taken to be

$p_b$	$p_{2A}$	$p_b$	$p_{2A}$	$p_b$	$p_{2A}$
1 or less	15.0	4	16.6	8	20.0
2	15.5	5	17.3	9	21.0
3	16.0	6	18.0	10	22.0
		7	19.0		

Of the important observed quantities in a test, the exhaust-pressure in a condensing steam-plant is the one which has usually been determined with the least accuracy, especially in the early work. Discussion of this point will be found in § 77 (b).

(c) EXPANSION OF THE STEAM.—The behavior of the steam both in general and in detail, can best be shown graphically, as by the representative diagrams in § 69. As a rough measure of the degree of expansion, the initial cut-off  $c$  and the total ratio of expansion  $r$  are here given, for all the cases where the original report contained the necessary data. The ratio thus expressed—see the definition on page 603—may differ quite a little from the true ratio of the final to the initial effective volume of the working steam which would be got by the method of § 19 (f), easily extended to the compound engine, and very conveniently applied directly to the combined diagram.

The general purpose of all the economy devices, especially compounding and steam-jacketing, is to make possible a full expansion of the steam in the engine, without the development of overwhelming losses on account of thermal interactions; but since the primary measurement of economy depends only on power and steam-consumption, we see that the form of the steam diagram has chiefly qualitative value, showing how and why certain results are secured, but not in itself making known the value of the results. These considerations reinforce the initial statement of this article as to the advantage of a diagram over the tabulation of measurements expressed numerically.

(d) MEAN PRESSURE.—The mean effective pressure  $p_m$ , referred to the L.P. piston in all the multiple-expansion engines, is chiefly



importance as one of the two factors determining the power of the engine; and in comparison with the initial pressure  $p_1$  it helps to indicate the amount of expansion that is realized. The first question to be settled in designing an engine which shall work under certain conditions is the value of this mean pressure. The examples in the tables vary quite widely, but it is fairly representative to say that most of the simple engines develop their normal power on from 30 to 40 lbs. M.E.P., while for the higher ratio engines the majority of values lie within the range from 20 to 25 lbs. In general, the transportation engines, both locomotive and marine, show higher mean pressures or a greater concentration of power. Further data on this point have been given in Tables 70 A to C, in connection with the subject of cylinder proportions.

(e) HORSE-POWER AND STEAM-CONSUMPTION.—These two quantities are overwhelmingly the most important determinations in a steam test; and their accurate measurement is less simple than may at first sight appear, and is by no means always realized. The indicator is not an instrument of precision, although with reasonable and proper care and skill it will give substantially correct results. The measurement of engine-power is most difficult when the load fluctuates, and it is always desirable to secure a steady and uniform load during the time of the test, if possible. The typical case of irregular fluctuation is found in a generator-engine supplying power for electric-railway service. As an example of the method which gives the best results under such conditions, we may cite the large engine in Test 259 Table C: here the effective power was measured electrically, and to it was added the combined engine and generator losses as found by "motoring" the unit with the engines empty. This same scheme can be used with turbines, when, however, it will include rotation losses also, besides the electrical and purely mechanical wastes of power.

As to the measurement of steam used, there is obviously more room for error in the feed-water method, and the tests must be of considerable length (at least four or five hours), in order to minimize the uncertainty as to the exact quantity of water held in the boiler. With a good surface-condenser the runs may be much shorter, say of one hour's duration. With lar                      complicated

engines there will be a number of subsidiary quantities to measure in the way of separator, receiver, and jacket-drains.

The magnitude of the steam-consumption can be better discussed after the thermodynamic efficiency has been developed as a criterion of performance, and this side of the subject will be found in Arts. (l) to (n).

(f) **QUALITY OF STEAM.**—The moisture-fraction  $m$  and the superheat  $t_s$  are given in the same column, since they cannot be

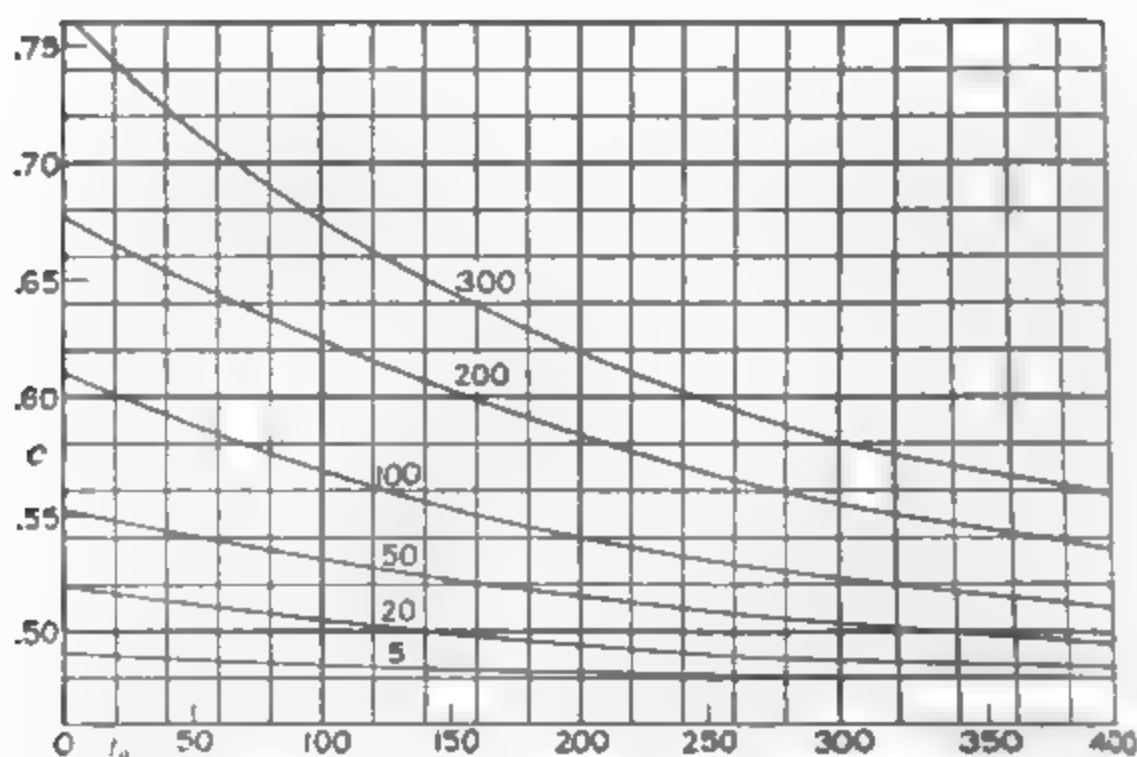


FIG. 685. Callendar's Curves for Specific Heat of Superheated Steam. Paper by Prof. H. L. Callendar in Phil. Trans. Roy. Soc., 1902.  $t_s$  = degrees of superheat, Fahr;  $c$  = specific heat under constant pressure. Each curve shows how  $c$  varies for the absolute steam-pressure marked on the curve.

coexistent. As to the former, quite one-half of the values are assumed, since so many of the tests were made before the steam calorimeter came into general use. In the matter of superheat the chief question is the amount of heat involved, given under conditions. To get this the writer has used Callendar's curves, reproduced in Fig. 685. The mean values of  $c$  for the given range and pressures are estimated from the diagram, with graphical interpolation. These are used in calculating  $h$  and  $w$ . It is not given in the table on account of lack of space, but can easily be found by dividing  $h$  by

*f.* Callendar's curves are by no means the last word on the subject of the specific heat of superheated steam, but are quite accurate enough for all practical purposes in the present connection. It will be noted that the writer's independent suggestion in Fig. 190 is along the same line.

(g) HEAT-CONSUMPTION.—The derivation of the heat of formation  $Q$  of one pound of the steam used in the engine is outlined in the definition, § 75 (d). The feed-water supplied to the boiler is made up chiefly of water at the ideal temperature  $t$ , belonging to exhaust steam of the pressure  $p_{2A}$ ; with this is mixed the hot water from the jackets and heaters. Thus in Test 117 the exhaust temperature is  $113.5^\circ$ ; of the pound of feed-water, .065 lb. is condensed steam from the jackets, coming supposedly at the full temperature,  $316.0^\circ$ . With water at  $113.5^\circ$ , the heat of formation  $Q_0$  of dry steam at 85.0 lbs. absolute pressure would be 1096.7 B.T.U. Now the jacket-water contains, per pound,  $(q_1 - q_2) = 287.0 - 81.6 = 205.4$  B.T.U. above water at  $113.5^\circ$ ; therefore the .065 lb. forming a part of each pound of feed-water brings  $.065 \times 205.4 = 13.3$  B.T.U., which we call  $q_H$ . The heat  $Q_0$  is diminished by this  $q_H$ , and also by the fraction  $m$  of the latent heat, or  $h$ , equal to  $.02 \times 891 = 17.8$ ; then  $Q = 1096.7 - 13.3 - 17.8 = 1065.7$ . The calculations are all made with the degree of numerical accuracy here indicated, but the last figure is dropped from the larger numbers in the table.

(h) TEMPERATURE OF FEED-WATER.—Thus to take the feed-temperature at its maximum attainable value makes the thermodynamic efficiency  $E_A$  belong to the engine alone, and not to the plant as a whole. With the very best heating appliances the water supplied to the boiler will be somewhat below the ideal temperature, the heat deficiency per pound ranging from as little as 5 B.T.U. with a non-condensing engine and open feed-heater to at least 15 B.T.U. with a condenser. In the latter case the temperature of the exhaust steam will always be less than that corresponding to the pressure, because this pressure is partly due to the air mixed with the steam; and further, the attainment of nearly the full temperature of the steam in the exhaust-pipe involves the use of a large surface which is rather exceptional.

Instead of  $t$  feed-temperature, which is essentially

the same as  $(t_2 + q_H)$ —numerical, not algebraic sum—we should come nearer to practical conditions by taking the best attainable temperature at from 5 to 20 degrees less, according to conditions. Then the value  $E_A$  got with the increased  $Q$  as divisor would be the plant-efficiency with the best possible feed-heating system: and any further increase in  $Q$  or drop in feed-water temperature would be chargeable to fault in this part of the plant, as distinct from the engine. But since the proper allowance in any case would be rather a matter of guesswork, it has seemed better to adhere to the scheme which has been defined and used.

The actual feed-temperature is lacking in many reports, and very often it was much lower during the test than in normal running on account of special arrangements for measuring the water; there was therefore not enough data to fill a column in the tables, but from the records of a number of good tests the conclusion has been drawn that in a condensing plant, where the supply is taken from the hot-well, the temperature of the water delivered to the feed-pipe is likely to be from 15 to 30 degrees less than the ideal maximum. Just as the condenser-discharge is quite a little cooler than the exhaust steam, so also is the jacket-discharge likely to be cooled by radiation, having somewhere near the same deficiency in heat-content per unit.

(i) REGENERATIVE FEED-HEATING.—An essential feature of the pure Rankine cycle is that the reception of heat from the fire to the water begins at the temperature of the engine-exhaust. When the engine has steam-jackets or reheaters the cycle is modified first by the withdrawal of a part of the steam from the working operation; secondly by the entrance into the exhaust of some of the heat from this steam. In other words, the jacket-steam rejects its latent heat into the cylinder-walls, its water-heat abstracted,  $t_2$ , into the feed-water, and only the heat below  $t_2$  into the exhaust. The underlying idea, of abstracting some heat from the working steam at high temperature (above the exhaust) and returning it to the feed-water with little or no drop, is most fully worked out in the scheme of regenerative feed-heaters; this has been quite fully outlined in § 69 (*h*) and is represented by Tests 365 and 366. The term “regenerative” comes from the analogy to the brick-

chambers in the Siemens furnace, which absorb and store heat from the escaping hot gases, then on the reversal of currents supply this heat to the entering cool gases. Here the heat is not stored, but is taken out of the temperature-lowering phase of the cycle on one side, and put right into the temperature-raising phase on the other side, with the least possible amount of drop.

The two tests just referred to, Nos. 365 and 366, differ from the others in that  $q_H$  is not calculated from an observed proportion of the jacket-steam, but is simply the difference in heat-content between feed-water at the actual temperature in the last heater and at the ideal exhaust-temperature  $t_2$ . These feed temperatures were  $311^\circ$  and  $334.5^\circ$  in the respective cases.

It must be clearly understood that this system can be advantageously applied only when the engine works under steady conditions, of load especially.

(j) PERFORMANCE PER POUND OF STEAM.—The energy-quantities  $W$ ,  $w$ , and  $W_H$  are sufficiently explained by the definitions in § 75 (d) and the reference in § 73 (a). It is here considered most simple and logical to take the pound of steam as the basal unit, as against the rather prevalent practice of calculating heat-quantities per horse-power and per unit of time, or for the whole plant per unit of time.

To make clear the methods used in getting the numbers in the tables, it may be well to work out an example or two.

EXAMPLE 1.—Test 117, already used in Art. (g) to illustrate the calculation of  $Q$ , has the following leading data:

$$p_{1A} = 85.0, p_{2A} = 1.4, S_H = 20.7, m = .02.$$

to find  $W = 2545 \div 20.7 = 123.0$  is very simple.

For  $w$  we use Eq. (430),

$$\Delta E \text{ or } w = -m_1 b_1 (t_1 - t_2):$$

from the Steam-tables IV. and XI.,

$$t_1 = 316.0, t_2 = 113.5, b_1 = 1.149;$$

and by substitution we get

$$w = -.02 \times 1.149 \times 202.5 = -4.65 \text{ B.T.U.}$$

To get  $W_n$  we interpolate in Table XII. to find  $E$ :

Take $E$ at $p_2 = 1.4$ , $p_1 = 80$ .....	258.53
Difference from $p_1 = 80$ to $p_1 = 90$ is .....	7.67
Add 0.5 of this to 258.53, getting .....	262.37

Combining  $\Delta E$  with  $E$  we have

$$W_n = 262.4 - 4.7 = 257.7 \text{ B.T.U.}$$

EXAMPLE 2.—In Test 134.2 the fundamental quantities are

$$p_{1A} = 159.5, p_{2A} = 0.71, S_n = 16.68, t_n = 227^\circ.$$

Then  $W = 2545 + 16.68 = 152.6$

For  $w$  we must now use Eqs. (431) to (433).

From Fig. 685 the specific heat  $c_1$  is taken to be 0.615.

Then the heat added is  $h = 0.615 \times 227 = 139.6 \text{ B.T.U.}$

The upper temperature  $t_1$  being  $363^\circ \text{ F.}$  or  $823^\circ \text{ AF.}$ , the entropy of superheating is

$$\begin{aligned} N_s &= 2.303 \times 0.615 \log \frac{823 + 227}{823} \\ &= 2.303 \times 0.615 \times 0.1058 \\ &= 0.1497 \end{aligned}$$

The rejection-temperature is  $T_2 = 91 + 460 = 551$ , and the heat rejection is

$$N_s T_2 = 0.150 \times 551 = 82.6 \text{ B.T.U.}$$

Subtracting this from the heat received we have

$$\Delta E \text{ or } w = 139.6 - 82.6 = 57.0 \text{ B.T.U.}$$

To get  $E$  from Table XII. we must now use double interpolation: producing the necessary part of the table, and inserting the differences, we have

$p_2$	150	$p_1$	160	
0.7	332.80	4.12	336.92	$p_2 = 0.71$ $p_1 = 159.5$
	6.22		6.19	
0.8	326.58	4.16	330.73	

Start with the smallest number . . . . .	326.58
0.71 is 0.9 of the way from 0.8 to 0.7,	
therefore add $0.9 \times 6.20$ . . . . .	5.58
159.5 is 0.95 of the way from 150 to 160	
therefore add $0.95 \times 4.12$ . . . . .	3.92
	<hr/>
The value sought is . . . . .	336.08

To this 336.1 add 57.0, getting 393.1 for  $W_R$ .

Note.—The examples are worked out largely with the slide-rule, so that the numbers may not be precise to the last figure.

(k) THE UNIT OF STEAM-QUANTITY.—With the idea of bringing the results of different engine-tests to a better condition for comparison, many engineers have adopted the scheme of reducing actual steam-consumption to equivalent dry steam, somewhat as in boiler-test the evaporation is always reduced to equivalent steam from and at 212°. The idea is, properly, to replace the actual steam (wet or superheated) by an amount of dry-saturated steam which will have the same heat-content: and the question at once arises, above what starting-point shall the steam-heat be measured? For engine-efficiency pure and simple, as in our tables, the initial state could be that of the ideal feed-water; for the plant efficiency, the actual feed-temperature must be used; and with different starting-points the reduction factor will be different. On account of this last consideration, the writer has preferred to give the actual weight of steam in every case, and to incorporate its quality into the calculation of the thermal quantities per pound. The occasional practice of simply subtracting moisture from total steam and calling the remainder the dry steam consumed by the engine is inexcusable.

(l) ABSOLUTE EFFICIENCY.—This quantity, the ratio of work put out to heat put in ( $E_A = W \div Q$ ), is the primary measure of the economy of the engine. Summarizing what is shown by the tables, without stopping at this point to study conditions, we make the following generalization:

## TYPE OF ENGINE

Simple engines, Table A . . . . .
Small high-speed, Nos. 211 to 216 . . . . .
Compound locomotives, Nos. 201 . . . . .
Large high-speed engines, Nos. 217 to 244 . . . . .
Compound Corliss-type engines with superheated steam and with modern heat, Nos. 251 to 257, 259, 281 to 286 . . . . .
Modern high-superheat compounds, Nos. 291, 292 . . . . .
High-ratio compounds and triples, Nos. 274, 301 to 316 . . . . .
High-grade pumping engines, Nos. 358 . . . . .
Good average marine engines . . . . .

The record is held by the regenerative engine, No. 366.

(m) **LIMIT OF PLANT-EFFICIENCY.**—By examination of realized performance, it is possible to estimate the limit of efficiency and illustrate the fact that the engine-efficiency is the upper limit of the steam-efficiency of the plant. If the engine-efficiency were to make the allowance of 15 to 30 per cent, the steam-efficiency would be increased by from 1.5 to 3 per cent, and the result would be the efficiency of the plant if the exhaust steam were used by auxiliaries.

When the essential pumps—condenser, feed-water, and air pumps—are driven by the main engine itself, there will be no loss of steam, but the effective power of the engine will be diminished by two or three per cent. Practical experience shows that this system is advantageous only when the locomotive is used for pumping plants.

The generally better, in fact the ideal, method is to exhaust steam from separate pumps to the main engine.



the latter is taken from the hot-well, and if the exhaust can all be condensed (in an open or mixing heater, and without raising the water up to  $212^{\circ}$ ), we have that all the heat coming from the boiler in this steam is returned, except the small amount spent in the effective work of the pumps. Provided only that there is enough margin in water-temperatures for this complete condensation (or not too much exhaust), we have a case where the power developed in the auxiliaries costs practically nothing: or, as a different form of statement, the auxiliary plant is now working at unit efficiency, because none of its rejected heat is wasted. With this simple circulation of heat—from the boiler, then nearly all back again—the auxiliaries have practically no influence upon the efficiency of the plant, which is determined wholly by the main engine. This arrangement is closely analogous to the working of an injector in boiler-feeding: but when we include condenser-pump as well as feed-pump, not quite all the energy goes back into the boiler.

If the steam from the pumps is wasted, as when these exhaust into the condenser, the ratio of steam to effective output is increased for the plant as a whole. Auxiliaries developing from 2 to 5 per cent. of the power of the main engine may easily use from 5 to 15 per cent. of the total steam. The larger values are for small plants with pumps in bad condition; and so great a proportion of steam could not all be utilized in feed-heating according to the scheme just described.

When it is impossible to heat the feed-water, as in a locomotive, the plant-efficiency is bound to be much lower than that of the engine alone.

(n) RELATIVE EFFICIENCY.—This is the most important criterion of engine-performance, since it takes account of the main determining conditions under which the thermodynamic operation is carried out, and shows how much margin there is between actual efficiency and the ideal maximum. It is by studying their effect upon this ratio that we can most intelligently estimate the value of the various schemes and expedients used to reduce the wastes in the engine. And it is, further, a very effective check upon over-high claims as to the results got in certain tests.

In the tables we see values ranging all the way from 0.35 (No.

331) to 0.85 (No. 115). Considering the range may divide this range as follows:

Rather poor performance . . . . .
Fairly good . . . . .
Very good . . . . .
Exceptional . . . . .

This applies to engines working at or near full load; with heavy overloads, and especially at low speeds, there will be a great falling off in efficiency. Below 0.40 will be invaded. Far down in this range are the pumps, which may use as much as 100 horse-power-hour. Of certain remarkable tests, those in tests Nos. 113, 115, 152, 224, 227, 304, and perhaps 344, are on the border-line, that they are too good to be true; while Nos. 113, 115, 152, 224, 227, 304, and perhaps 344, are on the border-line.

In Tests 365 and 366 the basis of comparison is the Carnot-cycle efficiency  $(T_1 - T_2)/T_1$ ,  $P$  and  $W_R$ . The writer is inclined to believe that the efficiencies shown in Test 366 are not to be taken as further confirmation, as by the tests of this

## § 77. Discussion of Engine Performance

(a) EFFECTS OF SIZE AND SPEED — When the size and in speed both make for losses, it is not easy to show a gradation in influences, because this requires a comparison with many elements of variation—such a comparison is clear and satisfactory than those that can be made when one element is varied in the working of the same engine. The size is perhaps best shown in the group of engines in the first half of Table C. A conspicuous feature is that it renders unnecessary special considerations of thermal losses, such as the steam-jacket, and simplifies and cheapens the engine.

The bad effect of very low speed, even when coupled with great size of cylinder, is strikingly shown by Fig. 686, which is of interest as one of the classic examples that first brought home to the minds of engineers and scientists the fact and the importance of cylinder-condensation. These diagrams are here all laid out to represent the performance of one pound of steam, with the same scales that are used in § 69, and with the constant-weight curve for the pound drawn in full line. The initial condensation is more than 50 per cent. for the earliest cut-off (at about one-eighth of the stroke); and we have here an unusually clear case of condensation continuing after cut-off, as indicated by the drop of the expansion-curve below

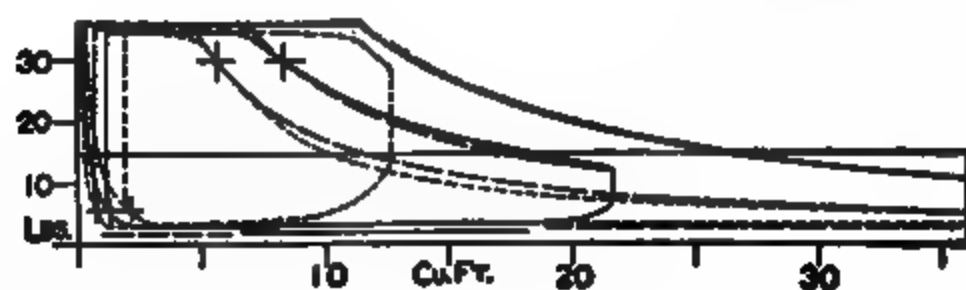


FIG. 686.—Diagrams from Isherwood's Tests of S.S. *Michigan*, Test 101.

the hyperbola. In showing an excessive thermal interaction these tests helped to make clear the fallacy of the early view that it was profitable to carry out a very high rate of expansion in a single cylinder, thus hastening the advent of the compound engine.

The most striking case of variation in the speed of an engine under test is seen in No. 141: while a very clear example of the effect of speed-change is given in the first four tests of the simple locomotive, No. 181, where  $E_R$  rises steadily with the speed. In compounds, Nos. 201 to 204, gain along this line is neutralized by increasing pressure-losses. The last test of No. 181 was made with the steam throttled, hence the lower efficiency there shown.

(b) PRESSURE-RANGE.—A very important fact that becomes evident by a study of the  $E_R$  column of Tables A and B is that the non-condensing engines develop a much larger proportion of their really possible performance than do those with vacuum exhaust—the difference being greater with simple than with compound engines. This is chiefly due to a greater cylinder-condensation on

account of larger temperature-ranges; but it does not pay to make the cylinder large. The large specific volumes of steam at low pressure make the cylinder very considerably "incomplete" as regards expansion. Figs. 597 to 608, and in the lower-ratio tests this effect is much more marked. Not advisable to make the (low-pressure) cylinder too large. The purely technical efficiency would be but little improved, as the frictional resistance will become too great for the steam-pressure.

The fact that a change in the exhaust pressure has a small influence upon the total efficiency  $E_A$  of the engine, as shown with its effect upon the ideal output  $W_1$ , is not true for great many of the earlier tests, less than has been paid to the accurate measurement of the exhaust pressure for some rather erratic variations in  $E_R$ . The small difference between the steam-turbine and the steam-engine in its total effect, is found in the steam-turbine can take very full advantage of the vacuum, but not equal to the engine at high-pressure. The importance of good vacuum very important in the steam-turbine leads to a careful measurement of the exhaust pressure.

(c) VARIATION OF EFFICIENCY WITH LOAD. The variation of this relation is one of the most important parts of the test, but to save space we have not given tables, preferring to illustrate this part of the test by curves. The first example, Fig. 687, according to this marine engine the speed varies with load irregularly as shown by the curve marked  $N$ . The basis for the curves, the I.H.P. measured at the engine, but that which would be developed at the engine and at the average (and uniform) speed of the engine. The ratio of expansion  $r$  varies as plotted. The curve for the steam-consumption,  $S_H$  (pounds per power-hour). The shape of this curve is similar to that of the steam-consumption with overload on account of incomplete expansion because of great thermal losses.

Typical curves covering a wider range of load are given in Fig. 8. The total-steam curve  $S$  (pounds per hour) is quite often plotted, since it gives rather a better idea of the performance at very light loads. If  $S_H$  were constant,  $S$  would follow a straight line towards the origin  $O$ . The point of tangency of the line  $OA$  with the curve of  $S$  corresponds with the minimum value of  $S_H$ . It is of interest to see how much (or how little)  $S$  rises above  $AO$  as the load falls off, since this is a criterion of the ability of the engine to carry a light load economically.

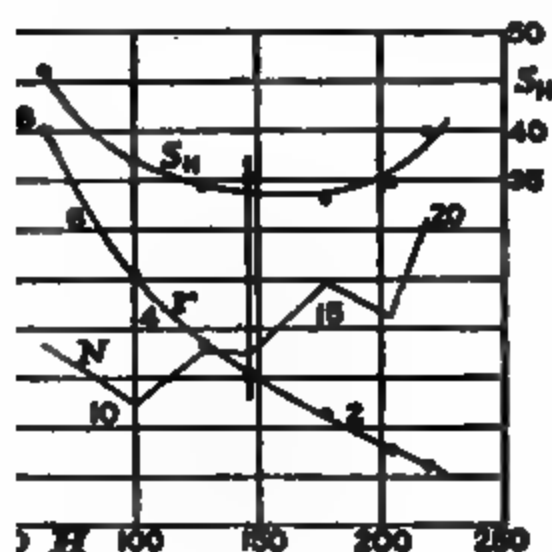


FIG. 687.—Curves for Tests of S.S. Michigan, Test 101. B. F. Isherwood, 1861.

$p_{1A} = 38$ ,  $p_{1A} = 2.1$ ,  $N = 15$ .

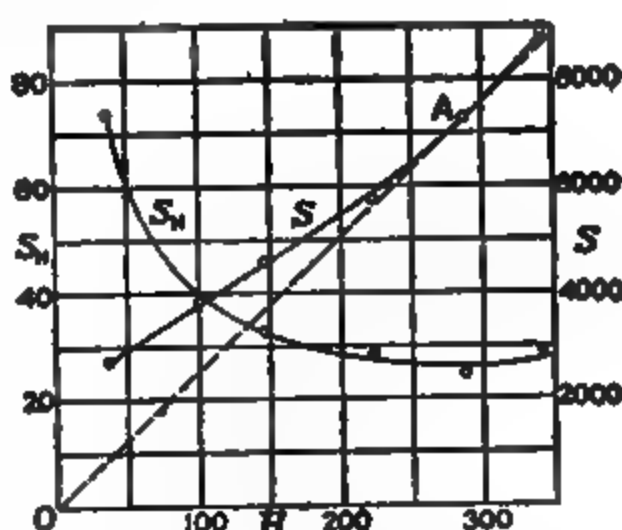


FIG. 688.—Simple Non-condensing Corliss Engine, 16x42. G. H. Barrus, 1893, E. R., July 22, '93-124.

$p_{1A} = 112$ ,  $p_{1A} = 14.7$ ,  $N = 86.5$ .

A very instructive lot of results are plotted in Fig. 689, of great interest as showing the effect of different degrees of compounding; marking the curve  $N$  and  $J$  distinguishes non-jacketed and jacketed operation. The principal curves show steam-consumption, of course, and we see how the best-load value of  $S_H$  is diminished by increasing the expansion; if the steam-pressure had been proportionately raised, the compound and triple would show up even better. Under neither of these latter conditions is the engine tried beyond what should be considered its rated load. Note how long is the range of power over which both compound and triple curves are nearly horizontal. The curves marked  $E$  show mechanical efficiency, and will receive comment in Art. (f).

Fig. 690 shows the working of a number of good engines. I. is noticeable thing about I. is the flatness of the  $S_H$  curve, or the fact that S points right toward the axis. The same rather unexpected variation is strongly shown in IV.; and the engine of Test 236 (here plotted) worked in the same manner. The kind of curve II. (similar to Fig. 688) is what is generally considered to be characteristic of the engine, and too little curvature throws doubt on the light-load measurements. A plausible explanation is this.

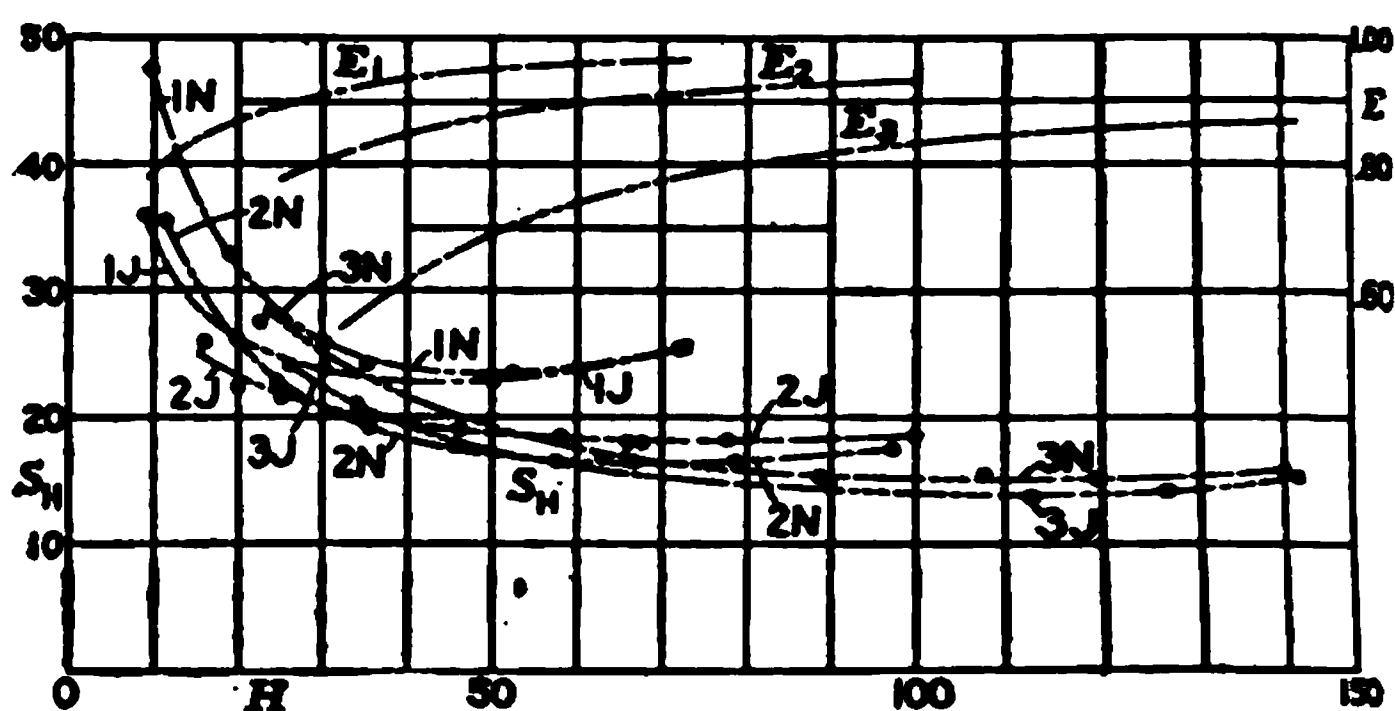


FIG. 689.—Tests of Experimental Corliss Engines at Sibley College, Cornell University—in Tables, Tests 112, 251, 301.

1 Simple	9	× 36	$p_{1A} = 115$	} $N = 87$ , Condensing
2 Compound	9, 16	× 36	$p_{1A} = 120$	
3 Triple	9, 16, 24	× 36	$p_{1A} = 130$	

When a compound engine has been running under a certain load and is then suddenly changed to a lighter load, the influence of the former condition persists for quite a while, the stored heat in the metal seeming to exert an influence entirely out of proportion to its actual amount. The three tests just referred to are all steady-state, surface-condenser tests; so that we may very properly question the reliability of the light-load determinations without resting upon the experimenters. The peculiar action just described was quite strongly brought out in the discussion, before the Soc. M. E., of the tests plotted in Fig. 689.

The curves numbered III. on Fig. 690 show exceptionally good results for so small an engine with saturated steam. In this case

the ideas of reducing the surface-area of the clearance-spaces and steam-jacketing all surfaces as far as possible are very fully carried out. The piston takes the form of two disks, so far apart from the rod that the space between them always remains in communication with a port which opens through the cylinder-wall from the jacket-space.

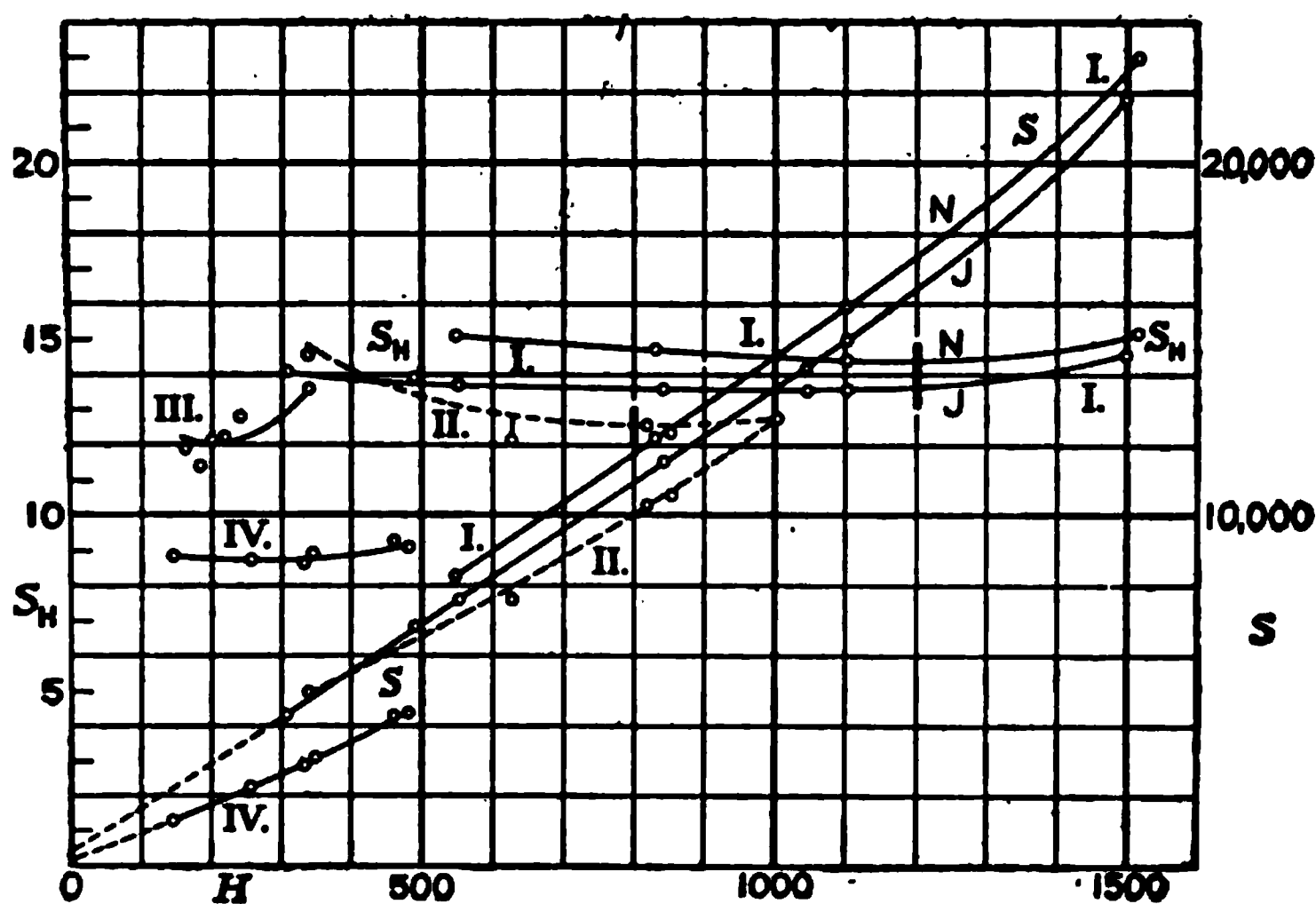


FIG. 690.—Tests of Compound Condensing Engines.

- I. Vertical McIntosh & Seymour, 23, 48×48; see Fig. 602.  
 $p_{1A} = 168$ ,  $N = 102$ . L. S. Marks, A. S. M. E., 25-443.
- I. Horizontal Corliss Engine, 20, 40×42; Test 257.  
 $p_{1A} = 165$ ,  $N = 121$ . I., II., Black lines show rated power.
- I. Cockerill-Francois Double-piston Engine, 14.8, 25.6×29.5.  
 $p_{1A} = 150$ ,  $N = 121$ . H. Hubert, 1904. ENG. 1905, II. 56
- V. Vertical High-superheat Engine, 21, 36×36; Test 292.  
 $p_{1A} = 132$ ,  $N = 101$ .  $t_s = 380$  to  $400^\circ$ .

The performance of a couple of very large generator engines is set forth in Fig. 691. The results are left in the terms in which they were measured,  $S_K$  being pounds of steam per kilowatt-hour. Since  $S_K$  includes both mechanical and electrical losses, increasing in relative amount at light loads, the curve of  $S_K$  should rise much more rapidly toward the left than would the curve of  $S_H$ . The curves

marked A and B for engine II. show the effect of changing the manner of variation of the receiver-pressure, or of the division of work between the cylinders. When running out into the overload region, raising the receiver-pressure will chiefly act to diminish  $S_K$  by receiver-drop, and this accounts for the better performance of case B. As on the last figure, the short heavy vertical lines mark rated load.

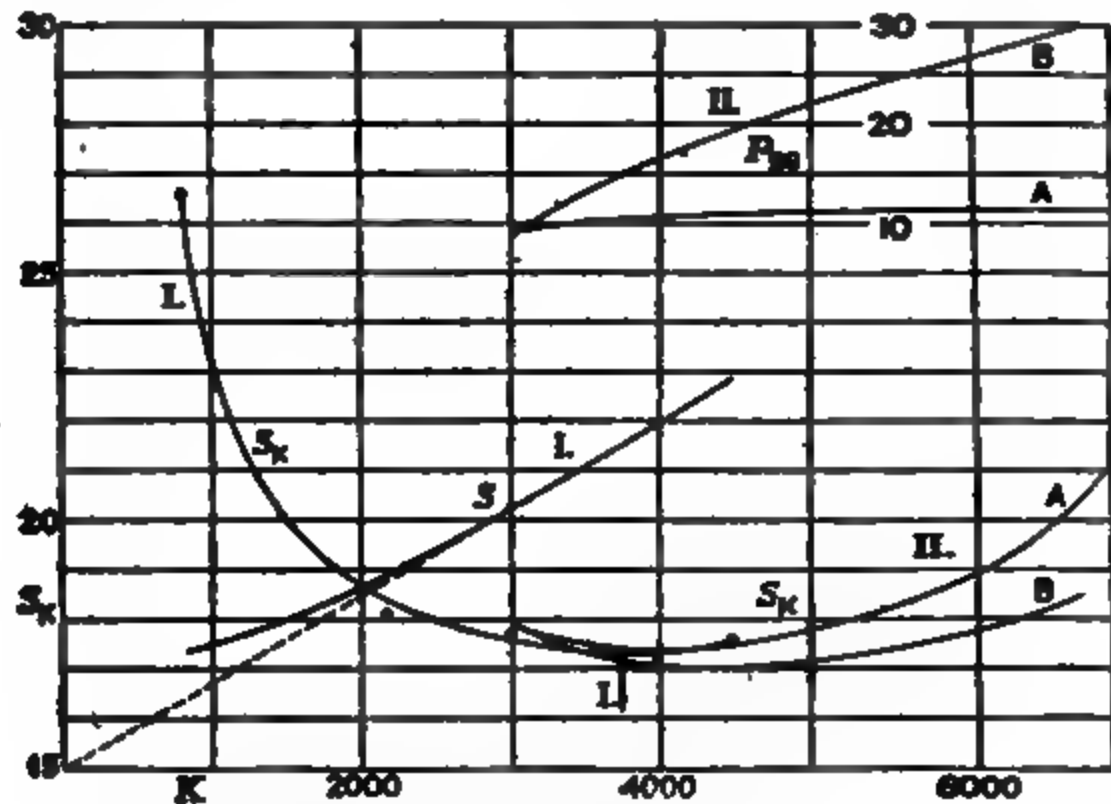


FIG. 691.—Very Large Compound Corliss Engines.

I. Three-cylinder Vertical, 44, 75.6–75.6×60; Test 271.

New York Edison Co., Waterside Station; Westinghouse Mach. Co.

II. Horizontal-vertical Duplex, 42, 86×60; Test 259.

Interborough (Subway) Power House, New York; Allis-Chalmers.

(d) EFFECT OF JACKETS AND HEATERS.—The evidence of the tabulated tests as to the degree of economy produced by these devices is decidedly variant. A summary is set forth in Table 77, where all the cases of experiment upon the same engine with and without jackets are collected and compared. Under  $N$  and  $J$  are given the steam-consumption per horse-power-hour,  $S_K$ , for the respective cases, while under "Gain" is the difference  $(N - J)$  expressed as a percentage of  $N$ ; a plus value means saving.



TABLE 77 A. EFFECT OF JACKETS AND HEATERS.

No.	N	J	Gain	No.	N	J	Gain
04	26.3	23.2	+12.7%	258	11.6	11.2	+3.6%
12	23.5	22.9	+ 2.5	271	12.1	12.7	-4.6
18	19.8	19.3	+ 2.5	274	11.3	11.5	-1.7
19	24.7	19.3	+21.9	301	15.3	13.9	+8.9
16 A	23.8	23.1	+ 2.9	304	10.4	10.4	-0.2
16 C	20.0	19.6	+ 1.7	332	16.9	16.6	+2.1
23	16.9	17.1	- 1.0	342	14.3	13.8	+3.0
43	14.4	13.6	+ 5.8	351	14.1	13.8	+1.8
51	16.5	18.0	- 9.1	371	23.2	21.3	+8.4
52	19.3	20.4	- 5.8				

jackets, a minus value, loss. The more recent tests, on larger and better engines, show but small differences. In this connection it will be well to review the statements made under Fig. 601.

An essential and quite important characteristic of jacket-action is well brought out by the curves in Fig. 689. At full load, the use or non-use of the jackets is largely a matter of indifference; but under light load, where the influences tending to cause cylinder-condensation are stronger, the jacket makes decidedly for economy. This leads us to the conclusion, which has been crystallized from general experience, that when, on account of small size of cylinder, low speed, early cut-off, or a big temperature-range, the cylinder-walls tend to be very active thermally, the jacket will be valuable; but under the contrary condition the jacket is less useful, and may even waste more steam than it saves. Fig. 689 gives further a conspicuous example of the contradictory evidence to be met with along this line: running simple or triple there is better performance with jackets and reheaters, running compound, without them—and there is no apparent reason for the difference.

The function of a reheater in the receiver is to furnish dry steam to the lower cylinder, and thereby diminish the tendency to initial condensation. This involves the re-evaporation of the moisture due to work done in the preceding stage, and unless there is ample heating-surface the re-evaporation may be incomplete, and so very little be gained. A thoroughly sound view of the principles involved is embodied in the scheme, applied in some of the best engines, exhaust from the higher cylinder through a sepa-

rator, thus removing most of the water drying and even superheating the steam in cylinder. Using boiler-steam to evaporate then work through only a part of the full is economical.

An incidental advantage of all these is the return of some heat to the boiler, discussion plant is so arranged that this hot water is sent to the boiler-feed, an unnecessary loss is incurred.

(c) SUPERHEATING.—In order to form a true economic value of superheating, we must consider increase in steam-consumption, and consider both absolute and relative. In applying this remember that if the engine can maintain superheat is added, there will be some gain in performance is better when a part of the heat temperatures above the boiling-point; if  $E_R$  increases more advantage gained.

The few tests which offer a chance for comparison gathered together in Table 77 B, where values under T and P for saturated and for superheat is the percentage of P over T. Other tests, Nos. 224, 228, 244, 292, and 304, show excellent performance with superheated steam, but there are no tests of steam which are in other respects enough like for comparison.

TABLE 77 B. EFFECT OF SUPER

No.	$t_b$	T	P	Gain	No.	$t_b$
102	96	078	086	+10.3%	282	129
134	227	136	122	-10.3	291	375
171	231	064	.115	+79.8	314	230
201-4	93	122	.134	+10.*	316	213

In only one case, No. 134, is there a lower temperature heat, while the greatest gain is, naturally, found

\* Here the seven tests under Nos. 201 to 203 are compared with under 204, and the means are compared.

engine like Nos. 171 and 172, where superheating is the only influence toward economy. The tests on larger engines appear to show an advantage of from 10 to 15 per cent. How great is the commercial value of this thermal gain is a question that is determined chiefly outside of the engine itself. The engine for superheated steam is likely to be the simpler in construction, since there is no logical place for the steam-jacket; the only expedient along this line is a reheater through the tubes of which the whole steam-supply is passed on its way to the first cylinder. The weakest point in a plant of this type will be the superheating surface, which is under much more trying conditions than that used for evaporation; it is therefore liable to rapid deterioration, and will have to be renewed at comparatively short intervals. And where a separately-fired superheater is used, there will be a larger proportional waste of fuel in banking fires than with the larger grates of the boiler proper.

In this latter connection, we may differentiate moderate superheating, up to say  $150^{\circ}$  F, which can be carried out in an apparatus forming a part of the heating-system of the boiler, and high superheat, up to  $300^{\circ}$  F., which requires a separate apparatus with its own fire. To put the superheater at a point in the hot-gas circulation of the boiler where the gases will be hot enough to raise steam  $200^{\circ}$  to  $300^{\circ}$  above the boiling-point will render these tubes liable to dangerous overheating if the boiler is pushed hard—although a good deal can be done to overcome this difficulty by using special dampers to divide the hot-gas current so that only a portion will pass through the superheater.

Taking account of extra cost of plant, both initial and for maintenance, it does not appear that the use of superheat offers much advantage over a well-designed engine with saturated steam.

(f) MECHANICAL EFFICIENCY.—In connection with the data presented in Tables 75 F and G, it will be well to refer to the general discussion in § 39. The brake horse-power, or the effective output of the engine alone, is given only in the "Power Engine" group of Table G, where we see normal-load values ranging from 0.82 to 0.96. A most instructive lot of results is partly tabulated in Nos. 112, 251, and 301, and more fully plotted in the efficiency curves

on Fig. 689. These data from the Sibley College engine show first of all how greater complexity of the machine increases the frictional loss, although the disproportionate increase due to the low-pressure section seems to indicate some local imperfection; another point of great interest is the small change in the friction power, under "Diff.", with variation of the load in any particular condition of operation.

The locomotive tests in Table G were all made with the dynamometer right back of the locomotive, instead of its being behind the tender as is unavoidable in a road test; the engine-friction is then increased only by the effect of the weight carried on the axes.

With the pumping-engine, the scheme generally used is to measure the pressure in the suction-pipe and in the discharge-pipe near the pump; to these is added the pressure due to the vertical "head" between the gages, and the result is the resistance against which the plunger works, and is used as the M.E.P. in calculating the pump horse-power. By this method the work of overcoming the water-friction is not included in the effective output of the pumping-engine, or this work is added to the machine-friction. Considering the further fact that in many cases the feed and condenser pumps are directly driven and their power-consumption included under "Diff.", we see that the large vertical pumping-engine is a remarkably efficient machine. The high mechanical performance in Nos. 331 to 337 would naturally be expected from engines of the Worthington type. The low efficiency in No. 354 is due to the fact that the power developed by two of the cylinders has to be transmitted through the crank-shaft, to help drive the one pump-plunger, which is tandem with the third cylinder.

The tests of combined engine and generator units in Table G indicate quite good engine efficiencies. The exceptional value in No. 271 can only be accounted for by the assumption that the indicator-measurements were not quite correct. And it may be remarked that too high thermal efficiency in Test 152 joins with the abnormally low mechanical efficiency to suggest a similar unreliability of the indicated power.

### § 78. Discussion of Turbine Tests.

(a) GENERAL CONSIDERATIONS.—In the compilation of Table 75 E, the aim has been to select good representative tests of the various types of turbines. In some cases, very little information is available beyond what is here directly represented; but for the more prominent types selection has been made from a large mass of data, the fuller presentation of which would only confirm the results here given. The general uniformity in kind of service and in working conditions produces a correspondingly small variation in the performance of turbines of any particular type.

Where a large series of tests has been reported from the works of the builders, those in charge of outside experts have been preferred. Of the references to Stodola, several are original, the others are thus given as most likely to be accessible to the reader. Much information along this line will be found in *Steam Turbine Engineering*, by Stevens and Hobart.

The computed values in this table are largely dependent upon the ideal steam per indicated horse-power-hour,  $S_H$ : this is based upon assumed mechanical and electrical efficiencies, as noted in § 75 (e), which are taken at what seem to be good probable values in order to make the thermodynamic results comparable with those of engine tests. Comparing Table E with F and G, we see that, as is reasonable, the turbine has been credited with higher mechanical efficiency than the engine: it was not intended that the rotation-losses—see § 73 (m)—should be included in the assumed  $E_M$ , but that they should rather be considered a part of the thermal waste. Then by so much as the turbine is considered more efficient mechanically when deriving  $S_H$ , by that much is a comparison based on  $E_A$  or  $E_R$  unduly favorable to the engine. Work delivered to the rotor per pound of steam is here emphasized rather than effective output: for the latter point of view a direct comparison of actual steam-consumption as given in Tables E and F, with due regard to variant conditions, is all that is required.

(b) *Single-stage Turbines*.—Of the De Laval tests, No. 401 is rather a special experiment, intended to prove the availability of very high superheat. The steam-consumption  $S_B$  for No. 402 is

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1. The first step is to identify the problem or question that needs to be answered. This involves understanding the context and the specific requirements of the task.

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So, the answer at  $T = 0$  is — The temperature of the air in the tube, the pressure and density are constant. The air is not stirred. The flow curve mathematically remains for the lifetime in general, that of  $S_2$  mathematically decreasing as the increase, not to an extent that will require the figure (see in Fig 62) at B, to be opened: if this is done the flow shows a hump.

Fig. 413 represents the low-pressure turbine which is fed by exhaust from non-reversing engines. Here the limiting pressure is below the control-value and at exhaust the pressure is placed at 1.5 bar, where a decided superheat is obtained.

turbine was run, not with exhaust steam through a Rateau "accumulator", but with steam reduced in pressure by throttling. Confirming what has been said in § 77 (b), these tests show a

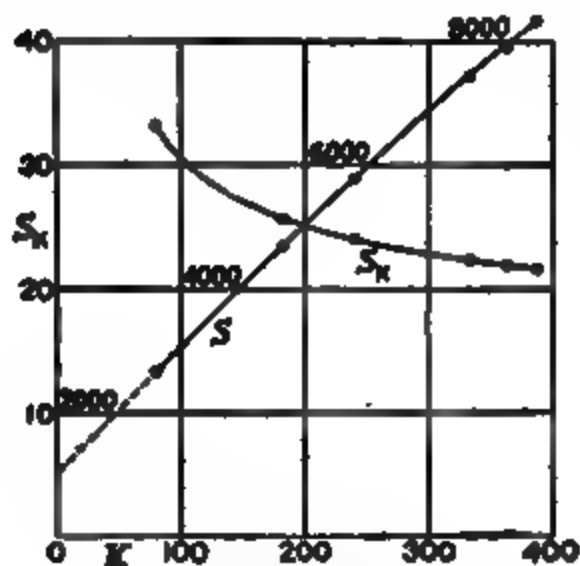


FIG. 694.—Zoelly Turbine, rated 500 H.P., Test 421.

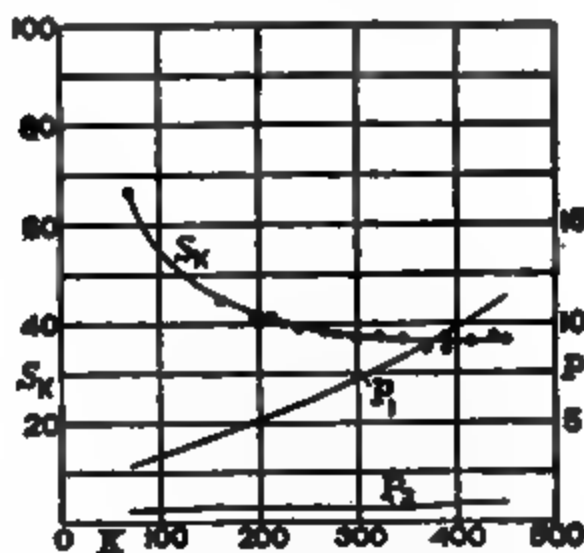
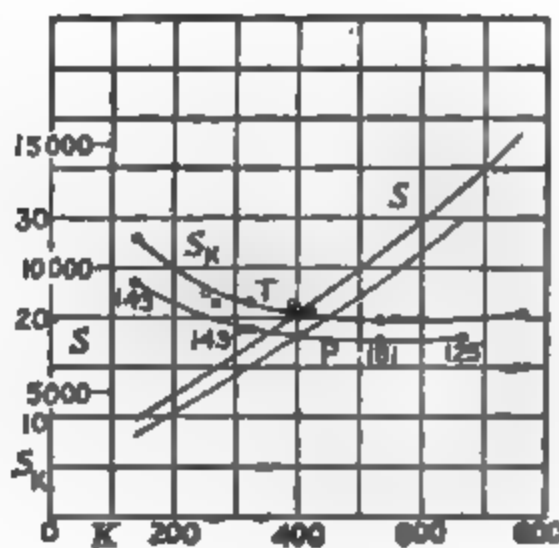


FIG. 695.—Low-pressure Rateau Turbine, rated 700 H.P., Test 414.

high efficiency in the utilization of the low pressure-range. The very reasonable suggestion has been made that a plant composed of high-pressure engines and low-pressure turbines would be more efficient than one with either type of prime-mover alone.

(d) THE CURTIS TURBINE.—Of this group of tests, No. 441 is plotted in Fig. 696 because it covers a wide range of load. Both saturated and superheated steam was used, the curves being marked T and P for these two cases; and along the  $S_K$  curve for superheated steam the degrees of superheat are marked. It will be noted that Test 441.2, with  $t_s=290^\circ$ , is not shown on Fig. 696. The relative efficiency keeps remarkably close to 60 per cent. for the whole group, taking an upward jump only at No. 445.



of steam is that it is essential to the work of the engine and the fact that it is not a perfect gas is a disadvantage that must be borne in mind. The fact that the steam is not a perfect gas is a disadvantage that must be borne in mind. The fact that the steam is not a perfect gas is a disadvantage that must be borne in mind.

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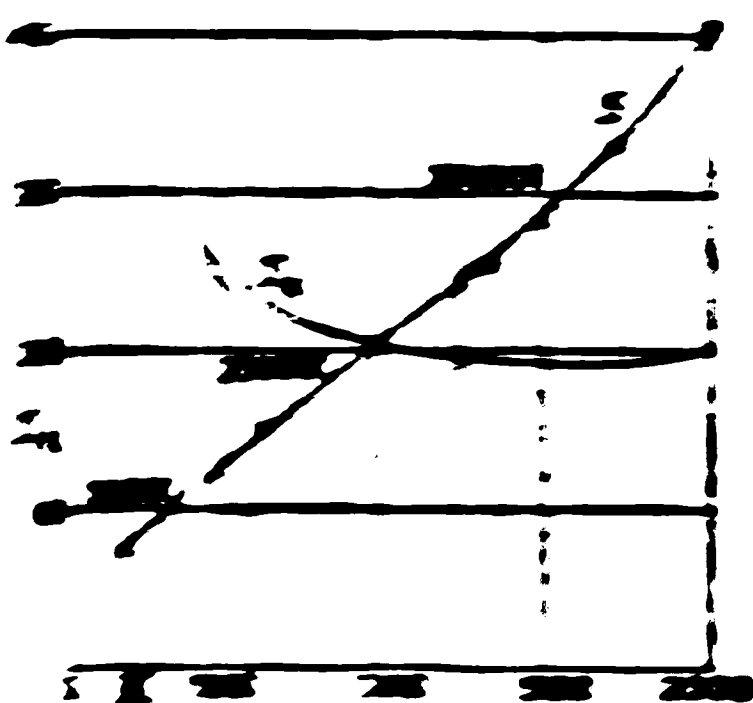


FIG. 67.—Temperature-Pressure Diagram, Test 442

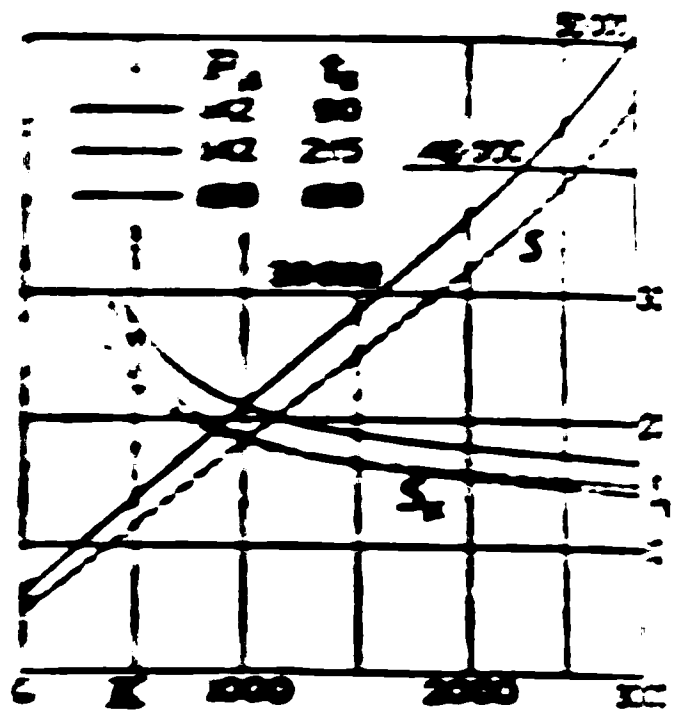


FIG. 68.—Pressure-Volume Diagram, Test 442

The best published performance by a Parsons turbine is that of Test 442, and that is shown in Fig. 67. Curves are drawn for different degrees of superheat and condensation pressure, as marked on the figure.

From the two tests in the Table, it may be said that the turbine is not small for high efficiency, while the other is not well suited for the first of the type.

**Effect of Vacuum.**—It is generally shown by tests that there is a steady gain in economy of steam with increasing vacuum. The examples in Table 75 E which offer a chance for comparison are entered in Table 78 A: Tests 442 and 443 are not really too far apart, but they are another effective pair, have really too much variation—in superheat and in condensation pressure.



as in vacuum. Differences in initial pressure must be considered in Nos. 409 and 411, so that a direct comparison of steam-summptions cannot well be made.

TABLE 78 A. EFFECT OF VACUUM.

	$p_{1A}$	$p_{2A}$	$S_{B, K}$	$t_2$	$E_A$	$E_R$
1.1	203	2.14	13.7 B	129	.156	.554
2	185	1.22	12.3	109	.171	.580
1.3	224	2.15	22.1 K	129	.144	.561
2	169	1.64	22.6	119	.143	.567
1.1	170	1.46	14.4 B	115	.166	.604
2	169	0.99	13.9	102	.170	.588
2.1	162	1.42	19.5 K	114	.171	.624
3	161	0.94	18.8	100	.175	.610
2.2	161	1.42	18.5 K	114	.173	.627
4	161	0.93	17.6	100	.179	.617

One thing made clear by even these few tests is that an advantage in steam used does not represent an equal gain in thermal performance:  $t_2$  is given to show how rapidly the ideal feed-temperature falls with the vacuum; and the effect of the resulting increase in the steam-heat  $Q$  is seen in the smaller relative difference between the values of  $E_A$  than between those of  $S_B$  or  $S_K$ . The production of extra good vacuum requires a very considerable increase in the condenser-equipment, which means that there will be more exhaust from the auxiliaries for heating the feed-water; but this extra steam helps to neutralize a gain in the main apparatus. It is safe to say that while the turbine can utilize low exhaust-pressure much better than can the engine, there is yet a limit below which it is not practically or commercially profitable to carry the reduction; and with some machines the limit may be higher than has been thought. In the last three examples of Table 78 A the falling off in  $E_R$  with decrease of  $p_2$  suggests that this limit may have been passed.

This brings us to the point that ability to utilize fully the lower

part of the pressure-range is partly a matter of the design and proportioning of the turbine; the last stages must be made big enough in sectional area of steam-channel and in size of vane, just as a multiple-expansion engine must have a very large L.P. cylinder to get the benefit of full expansion. An incidental advantage of the turbine is the rapid decrease of the steam-friction as the pressure is lowered—apparently the chief source of gain in the big single-stage turbine of No. 409—while in the engine internal friction relatively increases as the last cylinder is made larger. In both machines the low-pressure portion will act merely as a drag at light loads, and here again it seems that the advantage will lie with the turbine.

(g) EFFECT OF SUPERHEAT.—A number of comparisons of tests which were essentially the same except for the absence or presence of superheat are made in Table 78 B, all the examples being taken from Table 75 E. In terms of steam-consumption there is everywhere a decided saving; but when we transform to thermal economy the gain is much less. A small improvement in relative efficiency is

TABLE 78 B. EFFECT OF SUPERHEAT.

No.	$P_{1A}$	$P_{2A}$	$t_2$	$S_{2, K}$	$E_A$	$E_t$
401.1	99	14.5		39.5 B	.068	49
3	99	14.6	604	25.7	.079	51
404.1	221	1.67		15.5 B	.158	55
3	222	1.47	84	13.9	.164	57
421.1	159	1.03	7	21.5 K	.157	54
3	160	0.99	77	19.8	.163	56
441.1	165	1.0		19.8 K	.172	61
2	165	1.0	290	15.9	.185	66
451.2	169	0.99		13.9 B	.170	59
■	165	0.98	104	12.5	.179	61
4	168	0.99	180	11.5	.187	65
452.1	162	1.42		19.5 K	.171	63
2	161	1.42	76	18.5	.173	65
452.3	161	0.94		18.8 K	.175	63
■	161	0.93	78	17.6	.179	67

shown in every case, but the gain in absolute efficiency is not large enough to justify some of the rather high claims which have been made. That superheat is of more value in the turbine than in the engine is not substantiated by these tests, although the difficulties in the way of applying it are less than where there are valves and pistons. It is claimed further that steam-friction is diminished by superheating, and that the wear on vanes is less than with wet steam, and these advantages are decidedly useful.

A general review of the test tables, comparing steam-turbines with engines of good design, will show that the turbine has not yet quite come up to the engine in heat-economy. The one field in which the turbine has shown marked superiority is that of ship-propulsion. A close determination of the power of a marine turbine is impossible; but it appears to keep well up with the stationary turbine in economy, while the marine engine is far behind the best land engines.



# APPENDIX.

## TABLES FOR REFERENCE.

### TABLE XI.

#### SUPPLEMENT TO STEAM-TABLE.

THIS table covers very closely the range of pressure below atmosphere, furnishing conveniently and accurately numerical quantities which are of importance in the calculation of ideal steam-performance. The symbols are the same as in §§ 9 and 13 and in Table IV., namely:

- $p$  = absolute pressure, pounds per square inch;
  - $t$  = steam temperature or boiling-point, degrees Fahrenheit;
  - $q$  = heat of liquid, above 32° F.;
  - $H$  = total heat of steam, above 32° F.;
  - $s$  = specific volume, cubic feet per pound of steam;
  - $a$  = entropy of water-heating
  - $b$  = entropy of evaporation
- } see § 13 (c).

The volume  $s$  is calculated by means of the formula  $ps^{1.085} = 483$ —Eq. (61), § 10 (i)—and will be found not quite to agree with the values given in the first part of Table IV.

### TABLE XII.

#### PERFORMANCE OF THE RANKINE CYCLE WITH SATURATED STEAM.

The numbers in the body of the table give the output in B.T.U. from one pound of steam, when the cycle operates between the upper pressure  $p_1$  at the top of the table and the lower or exhaust pressure  $p_2$  at the side—both pressures being absolute and expressed in pounds per square inch. For the use of this quantity refer to § 73(a), § 75 (d), and § 76 (j). It is the same thing as  $AU'$  in Tables 16 A and B and  $E$  in Table 24 A and Table V.

To save the trouble of referring to the Steam-tables, values of the total heat  $H_1$  and of the feed-water heat  $q_2$  are here given, so that the heat received per pound of steam, which is

$$Q = H_1 - q_2,$$

can be easily obtained for any conditions. The ideal absolute efficiency is  $E \div Q$ .

TABLE XI.

PROPERTY

$p$	$t$	$g$	$H$	$r$
0.1	34.97	2.97	1092.60	1089.67
0.2	53.30	21.31	1098.20	1076.90
0.3	64.70	32.72	1101.67	1068.96
0.4	73.11	41.14	1104.24	1063.10
0.5	79.83	47.87	1106.29	1058.43
0.6	85.48	53.52	1108.01	1054.49
0.7	90.35	58.40	1109.50	1051.10
0.8	94.64	62.70	1110.81	1048.10
0.9	98.50	66.57	1111.98	1045.41
1.0	102.00	70.09	1113.05	1042.96
1.1	105.20	73.29	1114.03	1040.74
1.2	108.16	76.26	1114.93	1038.67
1.3	110.92	79.03	1115.77	1036.74
1.4	113.50	81.62	1116.57	1034.95
1.5	115.92	84.05	1117.30	1033.25
1.6	118.20	86.34	1117.99	1031.65
1.7	120.37	88.52	1118.65	1030.13
1.8	122.43	90.59	1119.28	1028.69
1.9	124.40	92.56	1119.88	1027.32
2.0	126.28	94.45	1120.46	1026.00
2.1	128.08	96.26	1121.00	1024.74
2.2	129.81	98.00	1121.53	1023.53
2.3	131.47	99.67	1122.04	1022.37
2.4	133.07	101.28	1122.53	1021.25
2.5	134.61	102.83	1123.00	1020.17
2.6	136.10	104.32	1123.45	1019.13
2.7	137.55	105.78	1123.89	1018.11
2.8	138.95	107.19	1124.32	1017.13
2.9	140.31	108.56	1124.73	1016.17
3.0	141.63	109.89	1125.14	1015.25
3.2	144.16	112.43	1125.91	1013.48
3.4	146.55	114.83	1126.64	1011.81
3.6	148.83	117.13	1127.33	1010.21
3.8	151.01	119.32	1128.00	1008.68
4.0	153.09	121.42	1128.63	1007.21
4.2	155.09	123.43	1129.24	1005.81
4.4	157.01	125.36	1129.83	1004.47
4.6	158.85	127.22	1130.39	1003.17
4.8	160.63	129.01	1130.93	1001.92
5.0	162.34	130.74	1131.45	1000.71
5.2	164.00	132.41	1131.96	999.55
5.4	165.60	134.02	1132.45	998.43
5.6	167.16	135.60	1132.92	997.33
5.8	168.68	137.13	1133.39	996.26
6.0	170.15	138.61	1133.84	995.23

LOW ATMOSPHERIC PRESSURE.				TABLE XI.			
	t	q	H	r	s	a	b
.2	171.57	140.05	1134.27	994.22	59.72	.25053	1.57420
.4	172.96	141.45	1134.69	993.24	57.96	.25274	1.56920
.6	174.31	142.81	1135.10	992.29	56.31	.25489	1.56436
.8	175.63	144.14	1135.51	991.36	54.76	.25699	1.55966
.0	176.92	145.44	1135.90	990.46	53.29	.25903	1.55509
.2	178.17	146.70	1136.28	989.58	51.90	.26101	1.55065
.4	179.40	147.94	1136.66	988.72	50.58	.26296	1.54632
.6	180.60	149.15	1137.02	987.88	49.33	.26485	1.54212
.8	181.77	150.33	1137.38	987.05	48.14	.26669	1.53801
.0	182.92	151.49	1137.73	986.24	47.01	.26849	1.53400
.2	184.05	152.63	1138.08	985.44	45.93	.27025	1.53007
.4	185.15	153.74	1138.40	984.66	44.90	.27199	1.52625
.6	186.23	154.83	1138.74	983.90	43.92	.27368	1.52252
.8	187.29	155.90	1139.06	983.16	42.98	.27534	1.51888
.0	188.33	156.95	1139.38	982.43	42.09	.27696	1.51533
.2	189.35	157.98	1139.69	981.71	41.23	.27854	1.51185
.4	190.35	159.00	1140.00	981.00	40.40	.28010	1.50840
.6	191.34	160.00	1140.30	980.30	39.61	.28164	1.50504
.8	192.31	160.98	1140.59	979.61	38.85	.28315	1.50176
.0	193.26	161.94	1140.89	978.95	38.12	.28462	1.49856
.2	194.19	162.88	1141.17	978.29	37.42	.28605	1.49542
.4	195.11	163.81	1141.45	977.64	36.74	.28747	1.49233
.6	196.02	164.73	1141.73	977.00	36.09	.28888	1.48929
.8	196.91	165.63	1142.00	976.37	35.46	.29025	1.48631
.0	197.79	166.52	1142.27	975.75	34.86	.29160	1.48338
.2	198.65	167.39	1142.53	975.14	34.27	.29293	1.48051
.4	199.50	168.25	1142.79	974.54	33.71	.29423	1.47769
.6	200.34	169.10	1143.04	973.94	33.16	.29552	1.47491
.8	201.17	169.94	1143.30	973.36	32.63	.29679	1.47218
.0	201.98	170.76	1143.54	972.78	32.12	.29804	1.46949
.2	202.78	171.57	1143.79	972.22	31.63	.29927	1.46686
.4	203.58	172.38	1144.03	971.65	31.15	.30047	1.46425
.6	204.37	173.18	1144.27	971.09	30.69	.30167	1.46168
.8	205.14	173.96	1144.51	970.55	30.24	.30285	1.45917
.0	205.90	174.73	1144.74	970.01	29.80	.30400	1.45669
.2	206.65	175.49	1144.97	969.48	29.37	.30514	1.45425
.4	207.40	176.25	1145.20	968.95	28.96	.30627	1.45183
.6	208.13	176.99	1145.42	968.43	28.56	.30739	1.44946
.8	208.85	177.72	1145.64	967.92	28.17	.30848	1.44714
.0	209.57	178.44	1145.86	967.42	27.80	.30957	1.44484
.2	210.28	179.15	1146.08	966.93	27.43	.31064	1.44257
.4	210.98	179.86	1146.29	966.43	27.07	.31170	1.44033
.5	211.33	180.21	1146.40	966.19	26.89	.31222	1.43922
.6	211.67	180.56	1146.50	965.94	26.72	.31273	1.43813
.7	212.00	180.90	1146.60	965.70	26.55	.31324	1.43705

TABLE XII.

OUTPUT

$P_1$	$P_2 = H_1 = Q_2$	OUTPUT		
		60 1171.2	70 1174.3	80 1177.6
0.1	3.0	361.9	370.9	379.8
0.2	21.3	355.2	341.6	349.7
0.3	32.7	314.3	323.7	331.9
0.4	41.1	301.2	310.7	318.9
0.5	47.9	290.8		308.7
0.6	53.5	282.1	291.7	300.1
0.7	58.4	274.6	284.3	292.8
0.8	62.7	268.1	277.9	286.4
0.9	66.6	262.3	272.1	280.6
1.0	70.1	257.0	266.9	275.4
1.2	76.3	247.8	257.7	266.4
1.4	81.6	239.9	249.8	258.5
1.6	86.3	232.9	242.9	251.7
1.8	90.6	226.7	236.8	245.5
2.0	94.5	221.1	231.2	240.0
2.2	98.0	215.9	226.1	235.0
2.4	101.3	211.2	221.4	230.3
2.6	104.3	206.8	217.1	226.0
2.8	107.2	202.7	213.0	221.9
3.0	109.9	198.8	209.2	218.1
3.2	112.4	195.2	205.6	214.6
3.4	114.8	191.8	202.2	211.2
3.6	117.1	188.6	199.0	208.0
3.8	119.3	185.5	195.9	205.0
4.0	121.4	182.5	193.0	202.1
4.5	126.3	175.7	186.2	195.3
5.0	130.7	169.5	180.1	189.2
5.5	134.8	163.9	174.5	183.7
6.0	138.6	158.6	169.3	178.5
7.0	145.4	149.3	160.0	169.3
8	151.5	141.0	151.8	161.2
9	157.0	133.7	144.5	154.0
10	161.9	127.0	137.9	147.4
11	166.5	120.9	131.8	141.4
12	170.8	115.3	126.3	135.9
13	174.7	110.0	121.1	130.7
14	178.4	105.1	116.3	125.9
14.7	180.9	101.9	113.1	122.8
15	181.9	100.6	111.7	121.4
16	185.3	96.2	107.5	117.2
18	191.4	88.2	99.5	109.3
20	197.0	81.0	92.4	102.3
22	202.2	74.4	85.8	95.7
24	207.0	68.3	79.8	89.7
26	211.5	62.6	74.2	84.2



POUND OF DRY SATURATED STEAM.								TABLE XII.
120	130	140	150	160	180	200	220	<i>p<sub>s</sub></i>
185.9	1187.8	1189.5	1191.2	1192.7	1195.7	1198.3	1200.8	
402.9	407.6	412.1	416.2	420.1	427.2	433.6	439.4	0.1
373.3	379.2	383.7	388.0	391.9	398.2	405.7	411.6	0.2
356.8	361.8	366.4	370.7	374.7	382.1	388.7	394.7	0.3
344.1	349.2	353.8	358.1	362.2	369.6	376.3	382.3	0.4
334.1	339.2	343.8	348.2	352.3	359.8	366.7	372.6	0.5
325.7	330.8	335.5	339.9	344.0	351.5	358.3	364.4	0.6
318.5	323.6	328.4	332.8	336.9	344.5	351.3	357.5	0.7
312.2	317.4	322.1	326.6	330.7	338.3	345.2	351.3	0.8
306.6	311.8	316.6	321.0	325.2	332.8	339.7	345.9	0.9
301.5	306.7	311.5	316.0	320.2	327.8	334.7	340.9	1.0
292.6	297.8	302.7	307.2	311.4	319.1	326.0	332.3	1.2
285.0	290.2	295.1	299.6	303.9	311.6	318.6	324.9	1.4
278.3	283.5	288.4	293.0	297.2	305.1	312.1	318.4	1.6
272.3	277.6	282.5	287.1	291.3	299.2	306.2	312.6	1.8
266.8	272.2	277.1	281.7	286.0	293.9	300.9	307.3	2.0
261.9	267.2	272.2	276.8	281.1	289.0	296.1	302.5	2.2
257.3	262.7	267.7	272.3	276.6	284.6	291.7	298.1	2.4
253.1	258.5	263.5	268.1	272.5	280.4	287.6	294.0	2.6
249.1	254.5	259.6	264.2	268.6	276.6	283.7	290.2	2.8
245.4	250.8	255.9	260.5	264.9	272.9	280.1	286.6	3.0
241.9	247.4	252.4	257.1	261.5	269.5	276.7	283.2	3.2
238.7	244.1	249.2	253.9	258.2	266.3	273.5	280.0	3.4
235.5	241.0	246.1	250.8	255.1	263.2	270.4	277.0	3.6
232.5	238.0	243.1	247.8	252.2	260.3	267.5	274.1	3.8
229.7	235.2	240.3	245.0	249.4	257.5	264.8	271.3	4.0
223.1	228.7	233.8	238.5	242.9	251.1	258.4	265.0	4.5
217.2	222.7	227.9	232.6	237.1	245.3	252.6	259.2	5.0
211.8	217.3	222.5	227.3	231.7	240.0	247.3	254.0	5.5
206.7	212.3	217.5	222.3	226.8	235.0	242.4	249.1	6.0
201.7	207.3	212.5	217.3	221.7	230.0	237.4	244.1	7.0
195.8	201.5	206.7	211.4	215.7	224.1	231.5	238.2	8
189.8	195.5	200.7	205.6	210.1	218.5	226.0	232.7	9
184.7	188.4	193.7	198.6	203.2	211.6	219.1	225.9	10
179.3	182.0	187.3	192.2	196.8	205.3	212.8	219.7	11
174.4	176.2	181.5	186.4	191.1	199.5	207.1	214.0	12
169.0	170.8	176.1	181.1	185.7	194.3	201.9	208.8	13
164.0	165.8	171.1	176.1	180.8	189.3	197.0	203.9	14
159.3	161.1	166.5	171.5	176.2	184.8	192.4	199.4	15
154.8	158.0	163.5	168.5	173.2	181.8	189.4	196.4	16
150.4	156.7	162.2	167.2	171.9	180.5	188.2	195.1	17
146.1	152.6	158.1	163.1	167.8	176.4	184.2	191.1	18
141.9	145.0	150.4	155.5	160.2	168.9	176.7	183.7	19
137.8	138.1	143.6	148.7	153.4	162.2	170.0	177.1	20
133.8	131.8	137.3	142.4	147.1	156.0	163.8	170.9	21
129.9	126.0	131.5	136.6	141.3	150.3	158.1	165.3	22
126.1	120.5	126.1			145.0	152.9	160.0	23

TABLE XII.—*Continued*

$P_1$ $P_2$ $H_1$	220 1200.8	240 1203.1	260 1205.3	280 1207.3	300 1209.3
0.1	439.4	444.8	449.5	454.2	458.5
0.2	411.6	417.1	422.1	426.7	431.1
0.3	394.7	400.2	405.3	410.0	414.4
0.4	382.3	387.9	393.0	397.8	402.2
0.5	372.6	378.2	383.4	388.1	392.6
0.6	364.4	370.1	375.3	380.1	384.6
0.7	357.5	363.2	368.3	373.2	377.7
0.8	351.3	357.1	362.3	367.1	371.7
0.9	345.9	351.6	356.9	361.7	366.3
1.0	340.9	346.7	352.0	356.9	361.5
1.2	332.3	338.1	343.4	348.3	353.0
1.4	324.9	330.7	336.1	341.0	345.7
1.6	318.4	324.3	329.6	334.6	339.2
1.8	312.6	318.5	323.8	328.8	333.5
2.0	307.3	313.2	318.6	323.6	328.3
2.2	302.5	308.5	313.9	318.9	323.6
2.4	298.1	304.1	309.5	314.5	319.3
2.6	294.0	300.0	305.4	310.5	315.2
2.8	290.2	296.2	301.6	306.7	311.5
3.0	286.6	292.6	298.1	303.1	307.9
3.2	283.2	289.2	294.7	299.8	304.6
3.4	280.0	286.1	291.6	296.7	301.5
3.6	277.0	283.0	288.6	293.7	298.5
3.8	274.1	280.2	285.7	290.8	295.6
4.0	271.3	277.4	283.0	288.1	292.9
4.5	265.0	271.1	276.7	281.8	286.7
5.0	259.2	265.3	270.9	276.1	281.0
5.5	254.0	260.1	265.7	270.9	275.8
6.0	249.1	255.3	260.9	266.1	271.0
7.0	240.4	246.6	252.3	257.5	262.5
8	232.7	239.0	244.7	250.0	254.9
9	225.9	232.2	237.9	243.2	248.2
10	219.7	226.0	231.7	237.1	242.1
11	214.0	220.4	226.1	231.5	236.5
12	208.8	215.2	221.0	226.3	231.4
13	203.9	210.3	216.1	221.5	226.6
14	199.4	205.8	211.6	217.1	222.2
14.7	196.4	202.8	208.7	214.2	219.2
15	195.1	201.6	207.4	212.9	218.0
16	191.1	197.6	203.5	208.9	214.1
18	183.7	190.2	196.1	201.6	206.8
20	177.1	183.6	189.6	195.0	200.2
22	170.9	177.5	183.4	189.0	194.2
24	165.3	171.9	177.9	183.4	188.7
26	160.0	166.6	172.6	178.2	183.5

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